PERFORMANCE RANKING OF FINNED TUBULAR HEAT EXCHANGER SURFACES

James Ernest Baskerville

PERFORMANCE RANKING OF

FINNED TUBULAR HEAT EXCHANGER SURFACES

by

JAMES ERNEST BASKERVILLE

B.S., UNITED STATES NAVAL ACADEMY
Annapolis, Maryland
(1969)

SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS
FOR THE DEGREE OF OCEAN ENGINEER AND MASTER OF SCIENCE
IN MECHANICAL ENGINEERING

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

May 1977

PERFORMANCE RANKING OF FINNED TUBULAR HEAT EXCHANGER SURFACES

by

JAMES ERNEST BASKERVILLE

Submitted to the Department of Ocean Engineering on May 12, 1977 in partial fulfillment of the requirements for the degree of Ocean Engineer and Master of Science in Mechanical Engineering.

ABSTRACT

The traditional method of data presentation for heat exchanger surfaces does not permit comparison of individual surface types in any simple manner. This data is most commonly presented in terms of heat transfer coefficients and friction factors referenced to the exposed area as a function of Reynolds number based on the minimum free flow area.

Soland [5] proposed a method of surface comparison for plate-finned heat exchanger surfaces in which the heat transfer coefficient and friction factor is referenced to the base area and Reynolds number is based on the open flow area, as though the enhanced surfaces were not present.

Theory and principle inherent in the Soland proposal are used to develop a method of surface comparison for finned tube surfaces as presented in (a) Kays and London [2], and (b) several supplemental surfaces furnished by the Trane Corporation. The derived comparison method is evaluated by application to a practical crossflow finned tubular heat exchanger design.

Appendix III provides a method for sizing crossflow finned tubular heat exchangers.

Thesis Supervisor: Professor Warren M. Rohsenow
Title: Professor of Mechanical Engineering

ACKNOWLEDGEMENTS

To my wife and children, an expression of "Thank You" for three years of thoughtful, loving understanding.

To Professor Rohsenow, a sincere feeling of appreciation for suggesting this thesis topic and for the many hours of guidance helping me through the "uncharted waters".

TABLE OF CONTENTS

TABLE OF CONTENTS (Cont'd.)

NOMENCLATURE

a_d — Minimum free flow dimension between tubes measured diagonally between adjacent rows from base to base as if fins were not present; see Figure 4, [FT]

a_t — Minimum free flow dimension between tubes measured transversely to the direction of flow from base to base as if fins were not present; see Figure 4, [FT]

A_b — Heat transfer area of the bare base tubular surfaces; equals length times heated perimeter times number of tubes; [FT^2]

A_c — Exchanger minimum free flow area, defined by (1a), [FT^2]

A_f — Constant from linearized equation for friction factor (f) of the form: $A\ RE^{B}$

A_F — Exchanger flow passage frontal area ignoring any enhancing surfaces; see Figure 4, [FT^2]

A_δ/A_T — Ratio of fin area to total transfer area, [FT^2]

A_{fr} — Exchanger total frontal area, [FT^2]

A_j — Constant from linearized equation for Colburn modulus (j) of the form: $A\ RE^{B}$

A_T — Total heat transfer area of exchanger on one side, defined by β times total volume, [FT^2]

b	Longitudinal center-to-center tube spacing, see Figure 4, [FT]
C	Flow stream capacity rate [WC_p], [BTU/HR-^{0}F]
C_D	Constant used in the computation of friction factor (f) for flow over smooth tube banks; defined by (32)
C_Δ	Constant required for the computation of radial fin efficiency; defined in Figure 5.
c_p	Specific heat at constant pressure, [BTU/LBM-^{0}F]
D_Δ	Constant required for the computation of radial fin efficiency; defined in Figure 5.
D_h	Hydraulic diameter, defined by (1a), [FT]
D_N	Nominal diameter, defined by (1b), [FT]
F	Transverse center-to-center tube spacing; see Figure 4, [FT]
f	friction factor based on total area (A_T); defined by (4a)
f_N	friction factor based on base area [A_b]; defined by (4b)
f_s	friction factor based on total area (A_T) for a smooth surface, defined by (31) or (32)
g_o	32.174 [LBM/LBF] [FT/SEC^2]

G_c	Mass flux based on minimum free flow area, $[A_c]$, defined by (2a), [LBM/HR-FT^2]
G_N	Mass flux based on free flow area, $[A_F]$, defined by (2b), [LBM/HR-FT^2]
h	Heat transfer coefficient based on total area, $[A_T]$ defined by (5a), [BTU/HR-FT^2-^{0}F]
h_N	Heat transfer coefficient based on base area, $[A_b]$; defined by (5b), [BTU/HR-FT^2-^{0}F]
j	Colburn j-factor based on total area, $[A_T]$; defined by (7a)
j_N	Colburn j-factor based on base area, $[A_b]$; defined by (7b)
j_s	Colburn j-factor based on total area, $[A_T]$, for a smooth surface, defined by (28)
K	Thermal conductivity [BTU/HR-FT-^{0}F]
ℓ	Fin height defined by r_e-r_o; for rectangular fin sheets, see Figure 5, [FT]
L	Flow length along the axis of the tubes (i.e. for the fluid inside the tubes); for fluid flowing normal to the tube banks, defined as N times b; [FT]
m	Component of fin efficiency $(\eta_{\delta-L})$; defined by [10]
N	Number of tubes

N_R	Symbol used in reference [2] for RE
Nu	Nusselt number; defined by (6a)
Nu_N	Nusselt number; defined by (6b)
NTU	Number of transfer units; defined by (21)
$\eta_{\delta-L}$	Fin temperature effectiveness of a longitudinal fin of rectangular profile; defined by (9)
$\Delta\eta_{corr}$	Correction factor applied to $\eta_{\delta-L}$ (9) to permit the use of the simplier [Tanh $m\ell$]/$m\ell$ equation for fin temperature effectiveness in lieu of Bessel Function type equations.
η_o	Total surface temperature effectiveness, defined by (8)
Pr	Prandtl Number [$\mu C_p/K$]
P	Pressure [LBF/FT^2]
P	Pumping Power [HP]
ΔP_f	Friction pressure drop [LBF/FT^2]
q	Heat transfer rate [BTU/HR]
q/A	Heat Flux [$BTU/HR\text{-}FT^2$]
RE	Reynolds number based on minimum free flow area, [A_c], defined by (3a), Labeled N_R by ref [2]

RE_N	Reynolds number based on free flow area, $[A_F]$, defined by (3b)
r_e	Radius of enhanced (finned) surface, [FT]
r_o	Outside radius of base tube, [FT]
r_h	Hydraulic radius, equals hydraulic diameter divided by four for a round tube, [FT]
T	Temperature, [^{0}F]
U	Overall heat transfer coefficient [BTU/HR-FT^2-^{0}F]
V	Volume [FT^3]
W	Mass flow rate, [LBM/HR]
X,Y,Z	Principle dimensions of heat exchanger, [FT]
X_D	Ratio of diagonal tube spacing to tube outside diameter
X_L	Ratio of longitudinal tube spacing to tube outside diameter, equals b divided by D_o
X_T	Ratio of transverse tube spacing to tube outside diameter, equals F divided by D_o

<u>Miscellaneous</u>

α	In tube geometry, angle defined by the longitudinal centerline and the diagonal centerline; see Figure

	7, [degrees]
β	Ratio of total transfer area on one side of the exchanger to total volume of the exchanger, labeled as α in reference [2], $[FT^2/FT^3]$
Δ_1	-0.00259, Constant used to calculate C_Δ
Δ_2	-0.00840, Constant used to calculate C_Δ
Δ_3	-0.02426, Constant used to calculate D_Δ
Δ_4	-0.04521, Constant used to calculate D_Δ
δ	Fin thickness, or average fin thickness if tapered, [FT]
ε	Exchanger effectiveness
μ	Viscosity [LBM/HR-FT]
ρ	Density $[LBM/FT^3]$

Subscripts

d	diagonal
e	enhanced surface
m	heat exchanger fin material
0	outside

Subscripts (continued)

os	one side
s	smooth surface
t	transverse
T	Total
2	Finned side, colder fluid flowing over finned tube banks
1	Tube side; hotter fluid flowing inside smooth tubes

LIST OF FIGURES

LIST OF FIGURES (Cont'd.)

LIST OF FIGURES (continued)

I. INTRODUCTION

A. Purpose

The objective of any heat exchanger comparison method is to enable the designer to select, from an often overwhelming number of enhanced surfaces, that unique surface which can be termed "optimum". The comparison technique should be in the form of a logical, easily implemented, inherently accurate procedure.

It should be recognized from the onset that regardless of the definition of "optimum" selected, the surface labeled as such may not, in fact, be the best for a given practical situation. Space limitations or cost restraints frequently restrict the designers flexibility. While a particular surface may be "optimum" from a least volume criteria, for example, it may well exceed the cost limitations or require an unacceptably long dimension. Violation of either of these practical constraints could well force the selection of an alternate surface.

Soland [5] proposed a method of comparison that permits performance comparisons of all of the Kays and London [2] plate finned type heat exchanger surfaces on four different bases:

a. Same exchanger shape and volume

b. Same exchanger volume and pumping power

c. Same pumping power and NTU

d. Same volume and NTU

Sheldon [6], (a) applied Soland's method to practical heat exchanger design problems; (b) evalueated the method effectiveness; and (c) performed a comparative analysis on plate finned surfaces in addition to those found in Kays and London.

The purpose of this paper is to:

1. adapt Soland's method for application to crossflow <u>finned tube</u> type surfaces presented in (a) Kays and London; and, (b) several supplemental surfaces furnished by Trane Corporation;

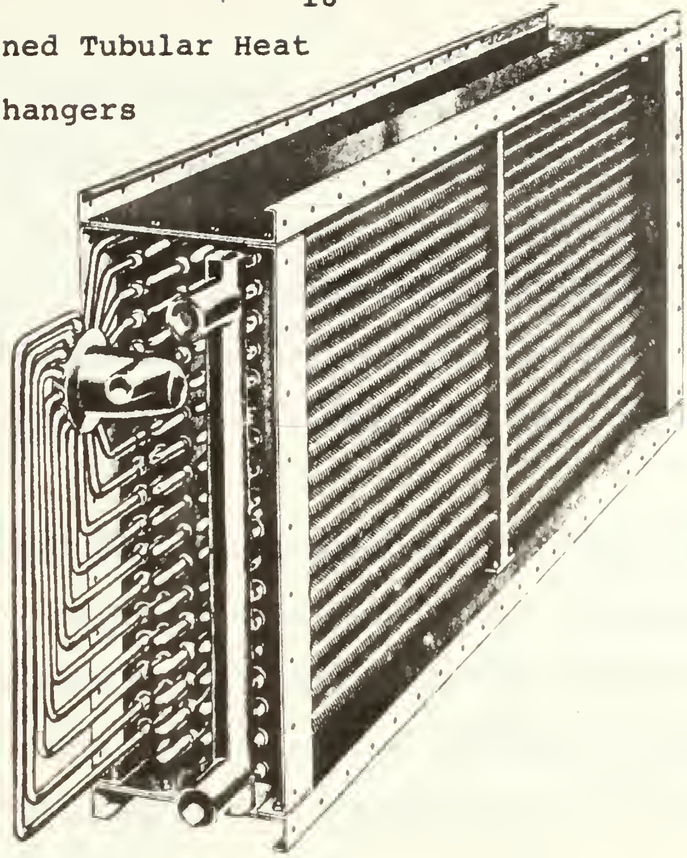
2. evaluate the method effectiveness by application to practical crossflow finned tubular heat exchanger design problems.

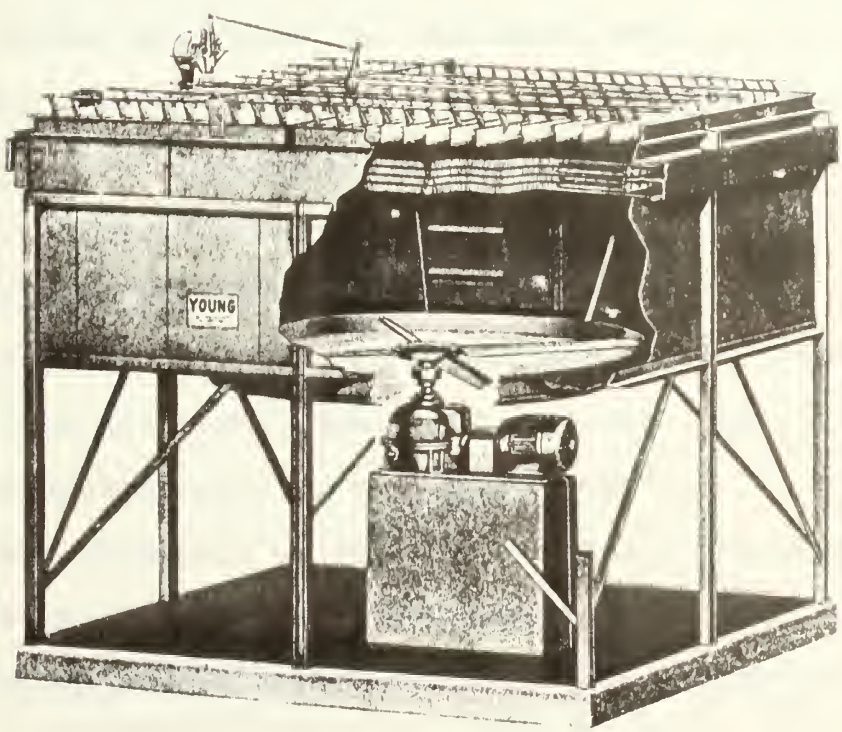
B. Background

Crossflow finned tubular heat exchangers have found application as gas turbine plant intercoolers, aircraft engine and electronics coolers, air conditioning units, heat exchangers, and numerous other uses. These applications most often involve gas-liquid service. [See Figure 1]

The performance of a given heat exchanger in which the ratio of the heat transfer coefficients between the two working fluids is greater than about three is markedly improved by the employemnt of enhanced surfaces. The increased surface area inherent with the employment of fins yields a net

FIGURE 1: Finned Tubular Heat Exchangers





improvement in heat transfer despite lower heat transfer coefficients. Finned tubular heat exchangers have surface-to-volume ratios (β) ranging between 20-200 ft^2/ft^3.

Reference [2] will be considered the primary source for data related to the design and testing of finned tubular surfaces, while supplemental data has been provided by the Trane Corporation.

Optimum design is often used synonomously with "lowest cost". Cost is frequently defined as (a) amortized acquisition cost related to the initial design and fabrication of the component, (b) operating costs, defined as amortized acquisition cost plus operating expenses and (c) cost to the system in terms of either weight or volume.

Consider aircraft engine component design, for example, where weight and volume are valuable quantities which can be directly equated to dollar cost. An unnecessarily large support component impacts either payload or endurance.

It is certainly conceivable that a heat exchanger could be designed with (a) the least acquisition cost, (b) the lowest projected operating expenses, and yet (c) have the highest weight or volume impact on the parent design and therefore by unacceptable to to "impact cost".

The estimation of dollar cost of a particular heat exchanger design is difficult to estimate, primarily due to the proprietary nature of the required cost estimating relations.

Companies that design and manufacture heat exchangers are often quite reluctant to release this data.

A method of dollar cost comparison will be proposed, but due to lack of data, will not be fully investigated.

The term "cost", therefore, as used in this paper, will most generally refer to "impact cost" and will serve as a means of relative ranking of proposed designs specifically by required volume, pumping power and NTU comparison.

C. Surface Testing and Data Presentation

In order to provide the required friction and heat transfer data for a specified finned tubular surface, experimental testing is required.

The experimental procedure employed by Kays and London is clearly outlined in reference [2].

Regardless of the technique, the results include fanning friction factor f, Prandtl number, Reynolds number, and Colburn j-factor. Figure 2 is an example showing the traditional method of data presentation.

D. The Task of the Designer

Assuming that the designer of a heat exchanger has defined such constraints as:

(a) allowable pressure loss,

(b) temperature change,

(c) amount of heat to be transferred,

FIGURE 2: Sample data plot showing traditional method of data Presentation

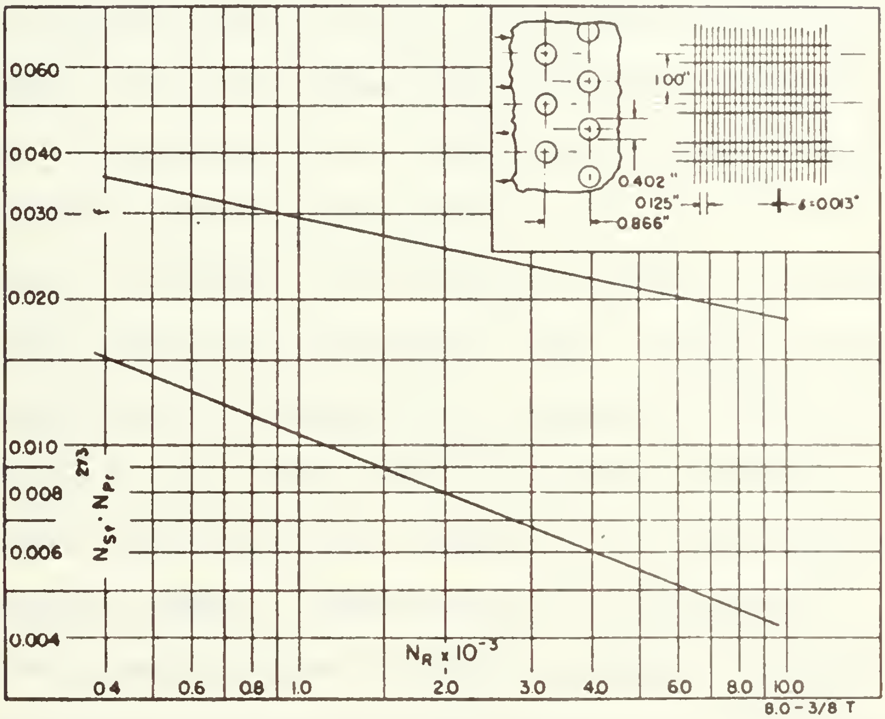


FIG. 100

FINNED CIRCULAR TUBES
SURFACE 8.0 - 3/8 T
(Data of Trane Co.)

Tube outside diameter - 0.402 in.
Fin pitch - 8.0 per inch
Flow passage hydraulic diameter - $4r_h$=0.01192 ft.
Fin thickness - 0.013 in.
Free-flow area/frontal area - σ =0.534
Heat transfer area/total volume - α =179 $ft.^2/ft.^3$
Fin area/total area - 0.839

Note: Minimum free-flow area in spaces transverse to flow.
These data are included in this compilation because they apply to a compact surface configuration of considerable technical interest for which no data have been obtained on this project.

(d) volume and weight restrictions,

(e) fouling and corrosion considerations

(f) materials

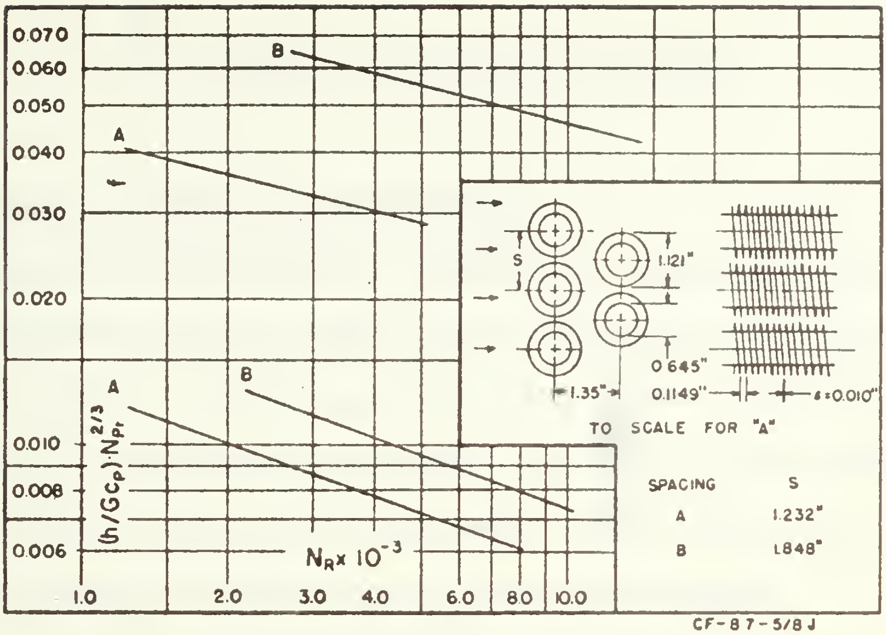
he is now ready to select the heat transfer surface(s) to be employed.

Having compiled a great number of candidate surfaces from (a) existing data, or (b) having designed and tested his own surface, he is now faced with the task of comparing these surfaces to determine which should be selected for the specific application under consideration.

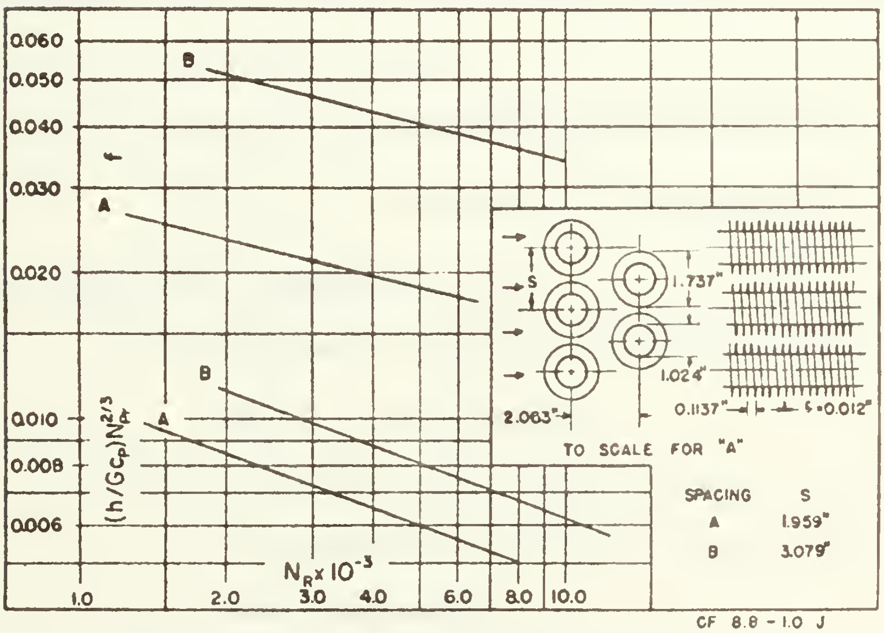
Figure 3 presents two examples from Ref [2] showing the traditional method of presenting f and j verses RE data . Note that in each case, Surface B has a superior heat transfer characteristic at a given Reynolds number; however, Surface A has a lower (and therefore, superior) friction factor at that same Reynolds number. Which surface is "optimum"?

There appears to be no obvious answer to this question and this fact has prompted several methods of surface comparison to be developed.

It is intended that the method presented herein will provide a solution to this dilemma.



Surfaces of Ref [2] Figure 97



Surfaces of Ref [2] Figure 99

FIGURE 3: Two examples from REF [2] showing the traditional method of presenting f and j verses RE surface data, and showing the difficulty in comparing surfaces A and B in each case.

II. PROPOSED COMPARISON TECHNIQUE

A. Derivation of Basis of Comparison

Soland, in reference [5], provides a detailed derivation of the proposed surface comparison technique used to analyze the performance of crossflow plate-finned heat exchangers.

Theory and principle inherent with the Soland method are used to derive a comparison technique to analyze the performance of crossflow finned tubular heat exchangers.

Comparison of the performance of various finned tubular surfaces assumes the following quantities are held constant:

A) W, flow rate

B) $T_{h,in}$, hot fluid inlet temperature

C) $T_{c,in}$, cold gas inlet temperature

Other important assumptions are:

E) The heat transfer resistance of the tube walls is negligible.

F) The controlling heat transfer resistance is assumed to be on the finned side of the heat exchanger.

Traditionally, data is presented in terms of heat transfer Colburn modulus, j, and friction factor, f, based on the total exposed area, A_T, as a function of Reynolds number, RE, based on minimum free flow area, A_c, and a hydraullic diameter,

D_h, of the flow passage.

The proposed comparison method converts these j and f magnitudes to new quantities j_N and f_N based on the heat transfer area of the bare base tube surface, A_b, and a Reynolds number, RE_N, which is referenced to the open flow passage area, A_F, as though the fins were not present. The effect of the fins is accounted for as an increased heat flux and hence larger h for for the bare base tube surface area. In order to incorporate the effect of the fins into j_N, the metal conductivity must be specified.

Table I shows the proposed new definitions and compares them with the analogous definition used by Kays and London [2]. Two cases must be considered as illustrated in Figure 4.

In order to convert data presented in the format of reference [2] to the new basis, various ratios are derived from basic definitions, equations (1) through (10), and Figure 4. These ratios are presented in Table II.

Two assumptions and an explanation of the calculation of radial fin efficiency are required in order to solve for the proper fin efficiency, η_δ, needed in the conversion relations:

1. The results presented herein will assume the gas flowing over the finned tube banks is air whose thermodynamic properties will be evaluated at 90°F.
2. Some of the results presented herein specify copper fins [K≈222 BTU/HR-FT-°F] and others specify Aluminum fins [K≈117.6 BTU/HR-FT-°F].

TABLE I. DEFINITIONS

QUANTITY	KAYS AND LONDON [2]		PROPOSED	
Hydraulic Diameter	$D_h \equiv \frac{4\ A_c\ bN}{A_T}$	(1a)	$D_N \equiv \frac{4\ A_F\ bN}{A_b}$	(1b)
Mass Velocity	$G_c \equiv \frac{W}{A_c}$	(2a)	$G_N \equiv \frac{W}{A_F}$	(2b)
Reynolds Number	$N_R \equiv \frac{G_c\ D_h}{\mu}$	(3a)	$RE_N \equiv \frac{G_N\ D_N}{\mu}$	(3b)
Friction Factor	$f \equiv \frac{\Delta P_f}{4\ \frac{bN}{D_h}\ \frac{G_c^2}{2\rho g_o}}$	(4a)	$f_N \equiv \frac{\Delta P_f}{4\ \frac{bN}{D_N}\ \frac{G_N^2}{2\rho g_o}}$	(4b)
Heat Transfer Coefficient	$h \equiv \frac{[q/\eta_o]A_T}{\Delta T}$	(5a)	$h_N \equiv \frac{[q/A_b]}{\Delta T}$	(5b)
Nusselt Number	$Nu \equiv \frac{D_h\ h}{k}$	(6a)	$Nu_N \equiv \frac{D_N\ h_N}{k}$	(6b)

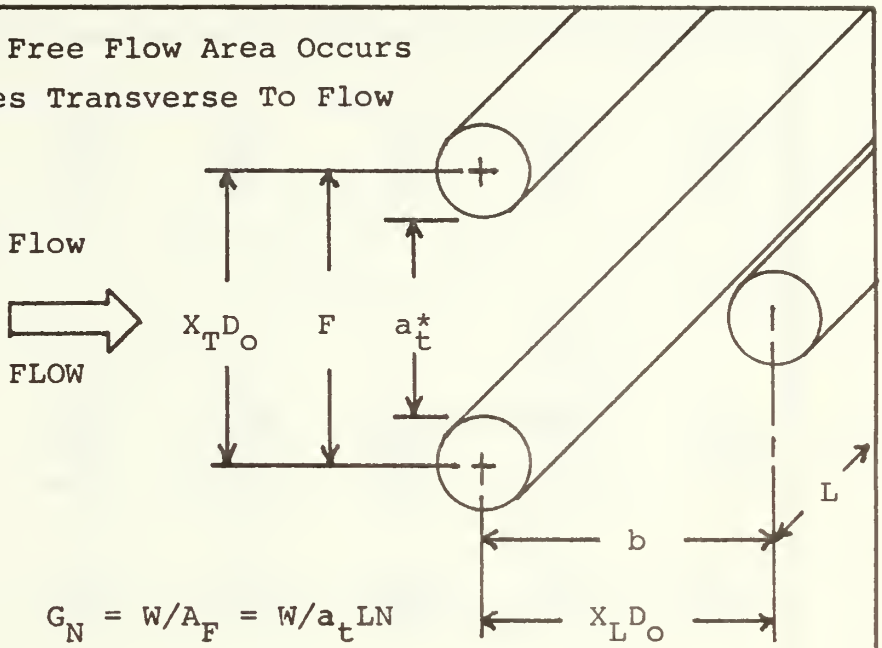
TABLE I. DEFINITIONS (continued)

QUANTITY	KAYS AND LONDON [2]		PROPOSED	
Colburn j	$j \equiv \frac{h}{G_c\ C_p} (Pr)^{2/3}$	(7a)	$j \equiv \frac{h_N}{G_N\ C_p} (Pr)^{2/3}$	(7b)
	$\eta_o \equiv 1 - \frac{A_f}{A_T}[1-\eta_f]$	(8)		
	$\eta_{f-L} \equiv \frac{\text{Tanh } m\ell}{m\ell}$	(9)**		
	$m \equiv \sqrt{(2h)/\delta\ k_m}$	(10)		

** Use of this formula for computing fin efficiency of a radial fin is permitted as outlined in Figure 5

CASE I: Minimum Free Flow Area Occurs In Spaces Transverse To Flow

*Minimum Free Flow Dimension



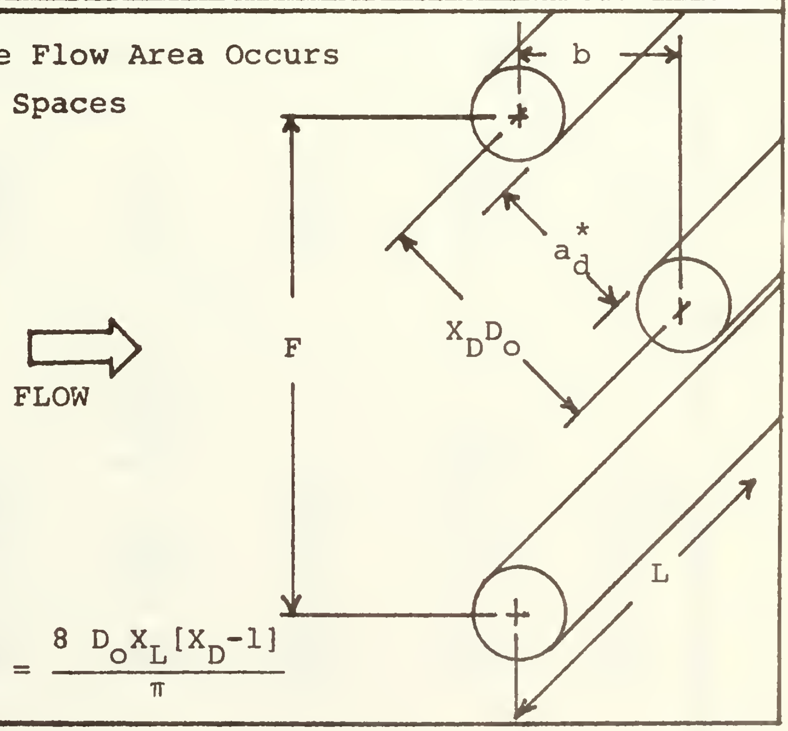
$A_F = a_t LN$

$A_b = \pi D_o LN$ $\qquad$ $G_N = W/A_F = W/a_t LN$

$V_T = FbLN$ $\qquad$ $D_N = \dfrac{4\ A_F b}{A_b} = \dfrac{4\ a_t b}{\pi\ D_o} = \dfrac{4\ D_o X_L [X_T - 1]}{\pi}$

CASE II: Minimum Free Flow Area Occurs In Diagonal Spaces

*Minimum Free Flow Dimension



$A_F = 2\ a_d LN$

$A_b = \pi D_o LN$

$V_T = FbLN$

$D_N = \dfrac{4\ A_F b}{A_b} = \dfrac{8\ a_d b}{\pi D_o} = \dfrac{8\ D_o X_L [X_D - 1]}{\pi}$

FIGURE 4: The Two Cases of Minimum Free Flow Area Occurance and the Associated Geometric Relations of Each Case.

TABLE II: Derived Ratios used to Convert Data Found in the Literature to the New Basis

CASE I		CASE II	
$\frac{A_b}{A_T} = \frac{\pi D_o}{\beta b F}$	(11a)	$\frac{A_b}{A_T} = \frac{\pi D_o}{\beta b F}$	(11b)
$\frac{A_F}{A_c} = \frac{4 a_t}{\beta F D_h}$	(12a)	$\frac{A_F}{A_c} = \frac{8 a_d}{\beta F D_h}$	(12b)
$\frac{G_N}{G_c} = \frac{A_c}{A_F} = \frac{\beta F D_h}{4 a_t}$	(13a)	$\frac{G_N}{G_c} = \frac{\beta F D_h}{8 a_d}$	(13b)
$\frac{RE_N}{N_R} = \frac{\beta b F}{\pi D_o}$	(14a)	$\frac{RE_N}{N_R} = \frac{\beta b F}{\pi D_o}$	(14b)
$\frac{f_N}{f} = \frac{16 a_t^2 D_N}{\beta^2 F^2 D_h^3}$	(15a)	$\frac{f_N}{f} = \frac{64 a_d^2 D_N}{\beta^2 F^2 D_h^3}$	(15b)
$\frac{j_N}{j} = \frac{\eta_o D_N}{D_h}$	(16a)	$\frac{j_N}{j} = \frac{\eta_o D_N}{D_h}$	(16b)

3. Figure 5 explains a proposed generalized procedure for estimating radial fin efficiency. This procedure provides a simplifed method of efficiency calculation in lieu of using more technically correct Bessel function type relations.

Figure 6 shows an example of data presented on both basis. It should be noted that additional curves of j_N verses RE_N would result for other magnitudes of fin thermal conductivity.

For any heat exchanger, the fluid pumping power per unit volume on one side is given by:

$$\frac{P}{V_{os}} = \frac{W \ \Delta P_f}{\rho \ V_{os}} \tag{17}$$

From equation 4b:

$$\Delta P_f = \frac{f_N \ 4 \ N \ b \ G_N^2}{D_N \ 2 \ \rho \ g_o}$$

From equation 2b:

$$W = G_N \ A_F$$

$$\frac{P}{V_{os}} = \frac{2 \ \mu^3 \ f_N \ RE_N^3 \ A_F \ b}{g_o \ \rho^2 \ D_N^4 \ V_{os}} \tag{18}$$

B

Compute fin efficiency assuming a Longitudinal fin of Rectangular Profile:

$$\eta_{f-L} = \frac{\text{Tanh } m\ell}{m\ell}$$

where:

$$m = \left[\frac{2\ h_2}{k_m \delta}\right]^{.50}$$

What geometric shape describes the fin being employed on the tube surface?

Radial Fin of Rectangular Profile

$$\Delta\eta_{correction} = C_\Delta - D_\Delta \ln(\eta_{f-L})$$

where:

$$C_\Delta = -.00259 - .0084 \ln \frac{r_o}{r_e}$$

$$D_\Delta = -.02426 - .4521 \ln \frac{r_o}{r_e}$$

[Ref 3 page 264]

Tapered (Hyperbolic) Radial fin

$$\Delta\eta_{correction} = 0$$

[Ref 4 page 3-116]

Rectangular Profile fin sheet [see below for ℓ defn]

$\Delta\eta_{correction} =$

Same as Radial Fin of Rectangular Profile using r_e^* (staggered bank)

$$r_e^* = \left[\frac{2\sqrt{3}}{\pi}\right]^{.50} \frac{X_d\ D_o}{2}$$

$$\ell = r_e^* - r_o$$

[Ref 4 page 3-116]

$$\eta_{f\text{-corrected}} = \eta_{f-L} - \Delta\eta_{correction}$$

Compute Surface η:

$$\eta_o = 1 - \frac{A_{fin}}{A_T}(1 - \eta_{f\text{-corrected}})$$

FIGURE 5: Computing fin efficiency of a Radial fin

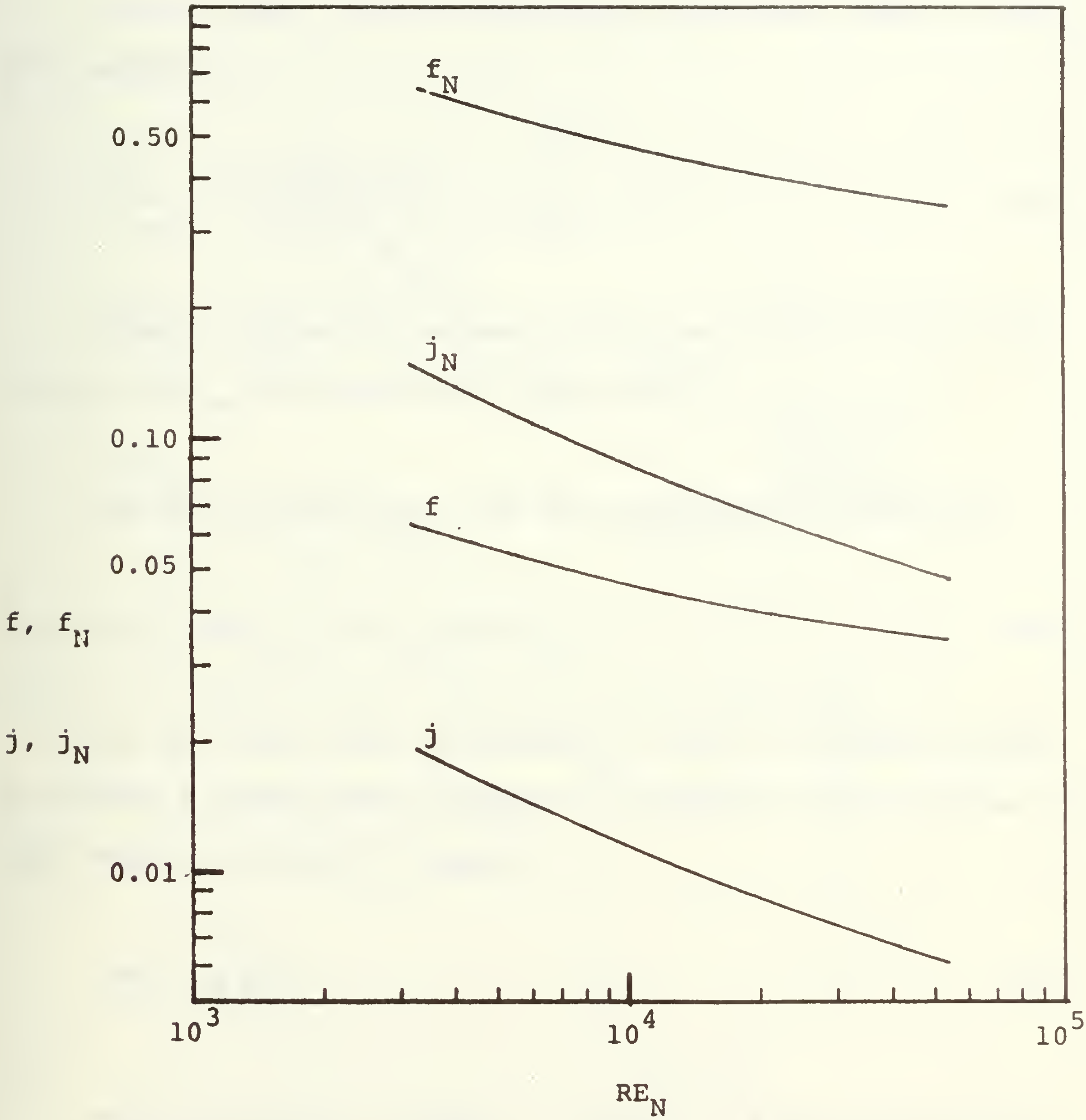


FIGURE 6: Comparison of Colburn j factors (j and j_N), and comparison of friction factors (f and f_N) for the Surface of Figure 94 [Ref 2].

For the same fluid at the same temperature level, μ and ρ are constant:

$$\frac{P}{V_{os}} \propto \frac{f_N \; RE_N^3 \; A_F \; b}{D_N^4 \; V_{os}} \tag{19}$$

Figure 7 shows the geometry involved in the calculation of $[A_F \; b]/V_{os}$ for both case I and case II.

The heat transfer for any heat exchanger is given by:

$$q = \varepsilon[T_{h,in} - T_{c,in}]W \; C_p \tag{20}$$

For any given flow arrangement, ε can be related to NTU by either a closed form equation or graphical curve similar to that shown in figure 8, where

$$NTU \equiv \frac{A \; h_N}{W \; C_p} \tag{21}$$

The relationship between ε and NTU is always monotonically increasing. In other words, an increase in $A \; h_N$ causes an increase in NTU which means ε and hence q are greater, given that the fluid properties and flow rate are held constant.

From equations (21) and (7b):

Case I: Minimum Free flow area Transverse to Flow Direction

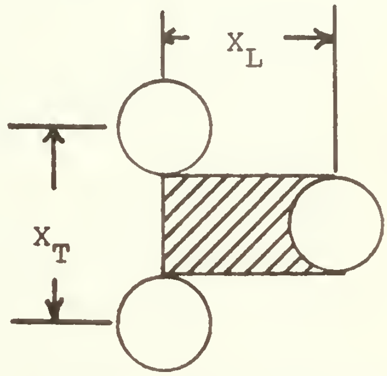
Total Volume on one Side:

$$V_{os} = [X_T D_o][X_L D_o]L - \frac{\pi}{4}D_o^2 L$$

L = dimension into paper along flow length of tubes.

"A_F b" Volume: ▨

$$A_F b = [X_T D_o - D_o] L X_L D_o$$



Ratio:

$$\frac{A_F b}{V_{os}} = \frac{[X_T - 1]X_L}{X_T X_L - \pi/4}$$

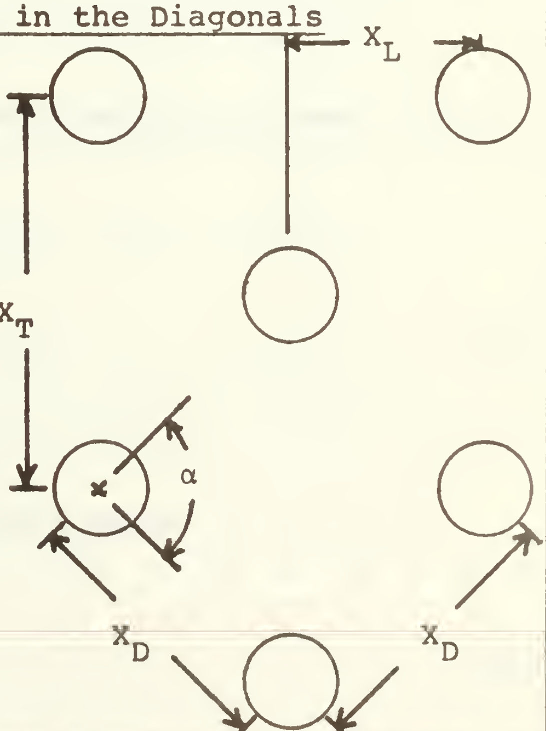
Case II: Minimum Free Flow area in the Diagonals

Total Volume on one Side:

$$V_{os} = \frac{[D_o^2 X_D^2 \sin\alpha - \frac{\pi}{4}D_o^2]}{2}$$

"A_F b" Volume:

$$A_F b = 2 X_L D_o [X_D - 1] D_o$$



Ratio:

$$\frac{A_F b}{V_{os}} = \frac{4[X_D - 1]X_L}{X_D^2 \sin\alpha - \frac{\pi}{4}}$$

FIGURE 7: Calculation of $[A_F b]/V_{os}$ for both case I and II.

$$\frac{NTU}{V_{os}} = \frac{A_b\ h_N}{V_{os}\ W\ C_p} = \frac{A_b\ j_N\ G_N\ C_p}{P_r^{2/3}\ V_{os}\ W\ C_p} \tag{22}$$

From equations (3b) and (1b):

$$\frac{NTU}{V_{os}} = \frac{4\ A_F\ b\ j_N\ RE_N\ \mu}{D_N^2\ P_r^{2/3}\ W\ V_{os}} \tag{23}$$

or

$$\frac{A\ h_N}{V_{os}} = \frac{4\ C_p\ \mu}{P_r^{2/3}}\ \frac{j_N\ RE_N}{D_N^2}\ \frac{A_F\ b}{V_{os}} \tag{24}$$

For the same fluid at the same temperature level, C_p, μ, and P_r are constant:

$$\frac{A\ h_N}{V_{os}} \propto \frac{j_N\ RE_N}{D_N^2}\ \frac{A_F\ b}{V_{os}} \tag{25}$$

And for the flow rate, W, also constant:

$$\frac{NTU}{V_{os}} \propto \frac{j_N\ RE_N\ \ A_F\ b}{D_N^2\ \ V_{os}} \tag{26}$$

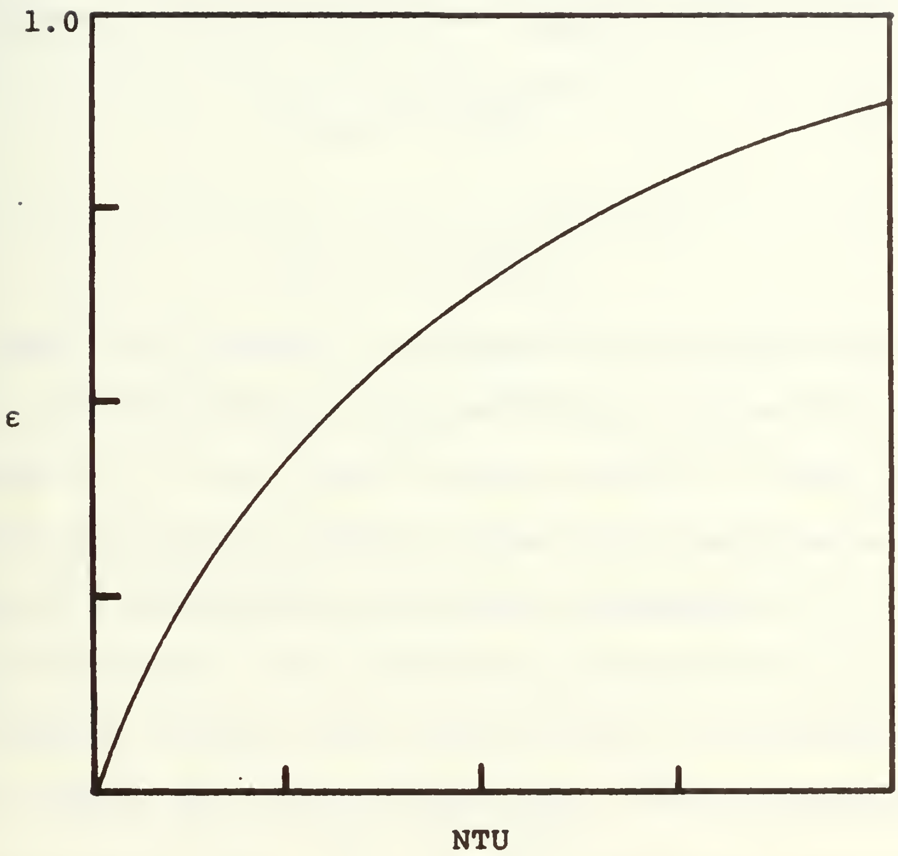


FIGURE 8: Typical Graphical Relationship between heat exchanger effectiveness and NTU. In general it is possible to express:

$$\varepsilon = \Phi[\mathrm{NTU},\ C_{min}/C_{max},\ \text{flow arrangement}]$$

Recognizing that the f and j verses RE plots for flow over finned tube banks from reference [2] can be closely approximated by a straight line on a log-log plot, implies that f and j can be represented by an equation of the form:

$$f_{(e)} = A_f \; RE^{B_f}$$

$$j_{(e)} = A_j \; RE^{B_j}$$

Table I of Appendix III lists the linearized equations of f and j for the surfaces represented by figures 92 -101 of reference [2]. This simplification allows for ease in programming the derived surface comparison equations as demonstrated in the procedure outlined in Appendix I.

In any event, the performance parameters may be calculated from enhanced surface data once the data is presented in the form f_N verses RE_N and j_N verses RE_N. This presentation may be (a) graphical due to the complex relations presented by f and j verses RE, as suggested by Soland or (b) analytical, as outlined in Appendix I, due in large part to the linear nature of f and j verses RE on a log-log plot.

B. Method of Surface Comparison

Having expressed the enhanced surface data in the form f_N verses RE_N and j_N verses RE_N, it is a relatively simple matter

to calculate the performance parameters of equations (19) and (26) and plot them. Figure 9 is an example of such a plot. The performance parameters of two hypothetical surfaces, labeled 1 and 2, have been plotted in order to demonstrate how a determination of heat exchanger relative performance may be made.

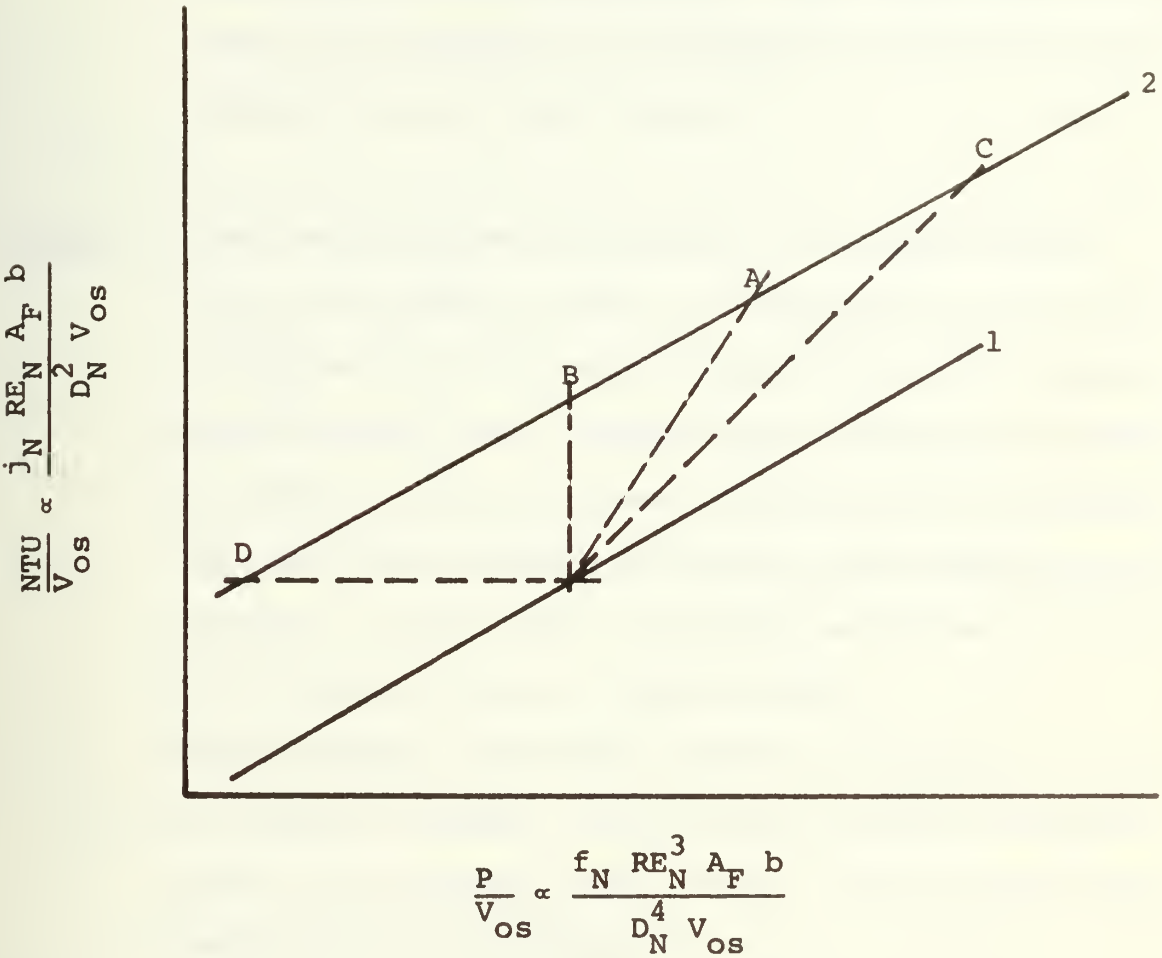
Four different comparisons are immediately available from figure 9 and are indicated by points a, b, c, and d on surface 2. Point o on surface 1 represents the reference heat exchanger design to which each of the four points on surface 2 will be compared.

Case A: Same heat exchanger shape and volume ($L_a = L_o$, $V_a = V_o$, $A_{F_a} = A_{F_o}$). Because W and A_F are fixed:

$$RE_{N_a} = RE_{N_o} \frac{[D_{N_a}]}{[D_{N_o}]} \qquad (27)$$

The results of this comparison are easily obtained as the ratios of ordinate values $[NTU/V_{os}]$ and abscissa values P/V_{os} of the two surfaces.

Case B: Same heat exchanger volume and pumping power ($V_b = V_o$, $P_b = P_o$). Point b is located on a vertical line through point o since the pumping power per unit volume is



<u>FIGURE 9</u>: Performance Parameter Curves For Two Hypothetical Surfaces Showing Points Used in Sample Comparisons.

Case B: (continued)

identical in both exchangers. The NTU ratio of the two heat exchangers is obtained as the ratio of the ordinate values from Figure 9.

Case C: Same pumping power and number of transfer units. ($P_c=P_o$, $NTU_c=NTU_o$) Point c is located on a line having slope equal to unity and through point o since both NTU and P are constant and each axis is inversely proportional to volume. The ratio of the volume required using surface 2 to the volume required using surface 1 is the ratio of either ordinates or abscissas at points o and c, respectively.

Since material costs are usually a function of volume or material weight, and thus heat transfer surface area, it is through Case C that a relative cost comparison may be attempted:

$$\frac{[\text{Total Cost}]_2}{[\text{Total Cost}]_1} = \frac{[\$/FT^2]_2\ [\beta]_2\ [V_T]_2}{[\$/FT^2]_1\ [\beta]_1\ [V_T]_1}$$

Note that as long as surface 2 lies above surface 1, the result will be a smaller volume required to do the same "job". [see Figure 9]

Case D: Same volume and number of transfer units. ($V_d=V_o$, $NTU_d=NTU_o$ or $q_d=q_o$). This case compares the pumping

Case D: (continued)

power required by surface 2 to that required by surface 1 for the case when both heat exchangers yield the same overall heat transfer performance. The comparison is made by taking the ratio of the abscissas (points o and d) in Figure 9, since a horizontal line on the performance plot has a constant value of NTU/V_{os}. The surface 2 Reynolds number for this case is the smallest of all four cases and consequently, the flow area is the largest. Since the volume is the same as the surface 1 exchanger, the length must decrease.

C. Summary

Using the equations derived herein it is possible to construct a performance parameter plot of NTU/V_{os} verses P/V_{os} (see Figure 9), from which it is possible to obtain useful performance comparisons between two heat exchangers for four different criteria.

III. COMPARISON OF FINNED TUBE SURFACES

A. General

Using the procedure derived in Chapter II and outlined in Appendix I, all the finned tube surfaces were plotted as performance curves of:

$$\frac{j_N \; RE_N \; A_F \; b}{D_N^2 \; V_{os}} \qquad \text{verses} \qquad \frac{f_N \; RE_N^3 \; A_F \; b}{D_N^4 \; V_{os}}$$

These performance curves assume constant W, $T_{c,in}$, and $T_{h,in}$, and are equivalent to NTU/V_{os} verses P/V_{os}.

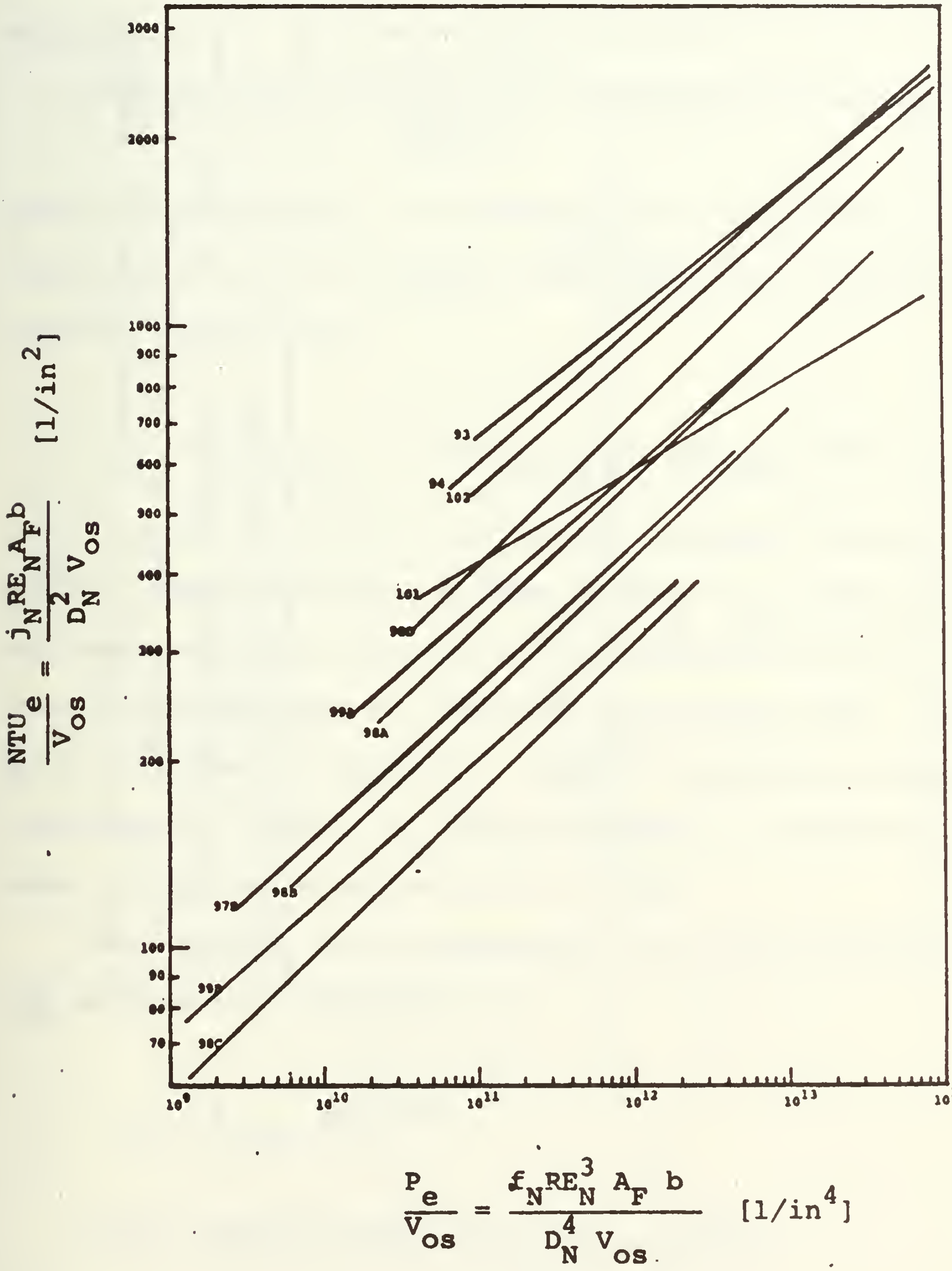
Figure 10 shows the resulting performance parameter plot presenting only the highest performance curve at each nominal diameter [D_N]. This study considered sixteen different surfaces having eleven different nominal diameters. Calculated performance results are presented for each surface considered in Appendix II.

B. Comparison to Smooth Surface

All of the sixteen finned tubular surfaces were then compared to a smooth surface having no enhancements (fins) and a nominal diameter identical to that of the enhanced surface.

PERFORMANCE PARAMETER CURVE FOR FINNED TUBULAR SURFACES

FIGURE 10



Note that subscript e will be used to denote the enhanced surface and the subscript s will be used to denote the smooth surface.

Smooth surface $\frac{j_s \ RE_N \ A_F \ b}{D_N^2 \ V_{os}}$ calculations were completed using the Zhukauskas [7] correlation for forced-convection, turbulent flow over staggered tube banks with the appropriate nominal diameter $[D_N]$.

$$j_s = 0.35 \ [RE]^{-0.396} \ [X_T/X_L]^{1/6} \ [D_o/D_N]^{-0.400} \qquad (28)$$

In order to justify the use of the above relation to predict smooth tube bank Colburn Modulus $[j_s]$, equation (28) was used to predict the actual heat transfer data for flow over bare tube banks as presented in reference [2]. The results are shown in figure 11. Maximum deviation between the experimental value of j_s (from reference 2) and the calculated value of j_s from equation (28) is ±7.5%.

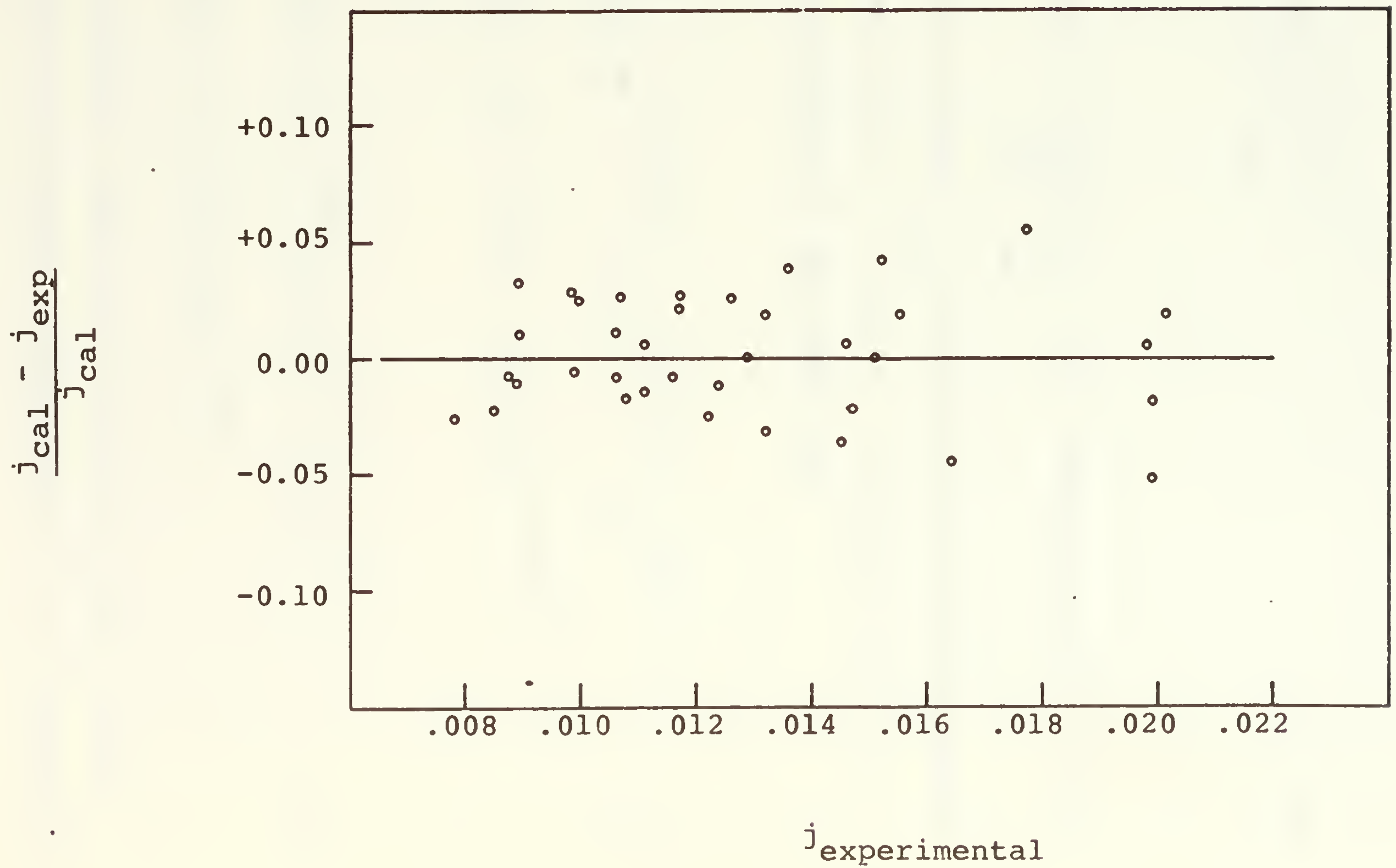
Note that j_s may be presented for each individual <u>enhanced</u> surface, in the form:

$$j_s = A_{j(s)} \ RE_N^{\ B_{j(s)}} \qquad (29)$$

For a smooth surface $h = h_N$, $G = G_N$, $RE = RE_N$

FIGURE 11

Deviation of the Zhukauskas [7] correlation for forced-convection, turbulent flow over staggered tube banks in predicting experimentally obtained value of Colburn modulus [j_s] as presented in reference [2].



Equation (26) may then be written for the enhanced surface as "<u>if smooth</u>":

$$\frac{NTU_s}{V_{os}} = \frac{j_{(s)} \; RE_N \; A_F \; b}{D_N^2 \quad V_{os}} \tag{30}$$

Smooth surface $\frac{f_s \; RE_N^3 \; A_F \; b}{D_N^4 \quad V_{os}}$ were completed using a correlation for friction factor (f_s) derived from first principles using the Kays and London [2] data for turbulent flow over staggered bare tube banks as the data base:

<u>Case I</u>:

$$f_s = 0.7184 \; RE^{-0.2407} \; [X_T-1]^{0.703} \; [X_L]^{C_T} \tag{31}$$

$$\text{where:} \quad C_T = 0.828 - 16.596 \; RE^{-0.4277}$$

<u>Case II</u>:

$$f_s = 1.0000 \; RE^{-0.1892} \; [X_D-1]^{0.703} \; [X_L]^{C_D} \tag{32}$$

$$\text{where:} \; C_D = 130.10 \; RE^{-0.7284}$$

Reference to Case I and Case II differentiates between minimum free flow area occuring transversely (I) or diagonally

(II) to the flow direction. (see Figure 4).

In order to justify the use of equations (31) and (32) to predict smooth tube bank friction factor (f_s), these relations were used to predict the actual experimental friction factor data for flow over bare tube banks as presented in reference [2]. The results are shown in Figure 12. Maximum deviation between the experimental value of f_s (from reference 2) and the calculated value from either (31) or (32) is ≈3.0%.

The friction factor, for each individual enhanced surface may be presented in the form:

$$f_s = A_{f(s)} \, RE_N^{B_{f(s)}} \qquad (33)$$

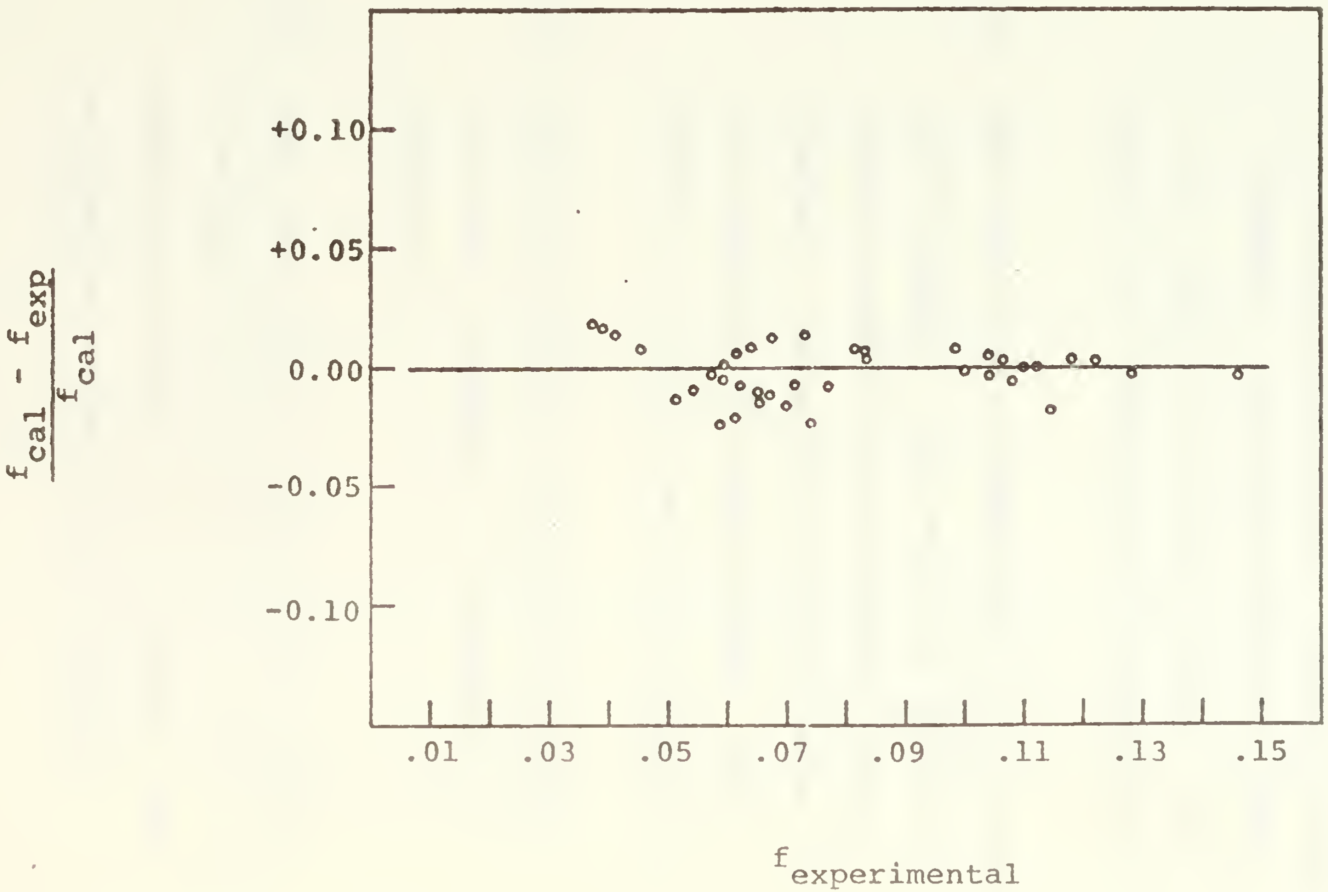
Equation (19) may then be written for the enhanced surface as "if smooth":

$$\frac{P}{V_{os}} = \frac{f_s \, RE_N^3 \, A_F \, b}{D_N^4 \, V_{os}} \qquad (34)$$

Of the sixteen surfaces investigated, five have a nominal diameter $[D_N] \approx 1.60$ inches. Since this offers an excellent means of comparison, cases A,B,C, and D where calculated for these five surfaces:

FIGURE 12

Deviation of the derived correlation for forced-convection, turbulent flow over staggered tube banks in predicting experimentally obtained value of friction factor [f_s] as presented in Ref [2].



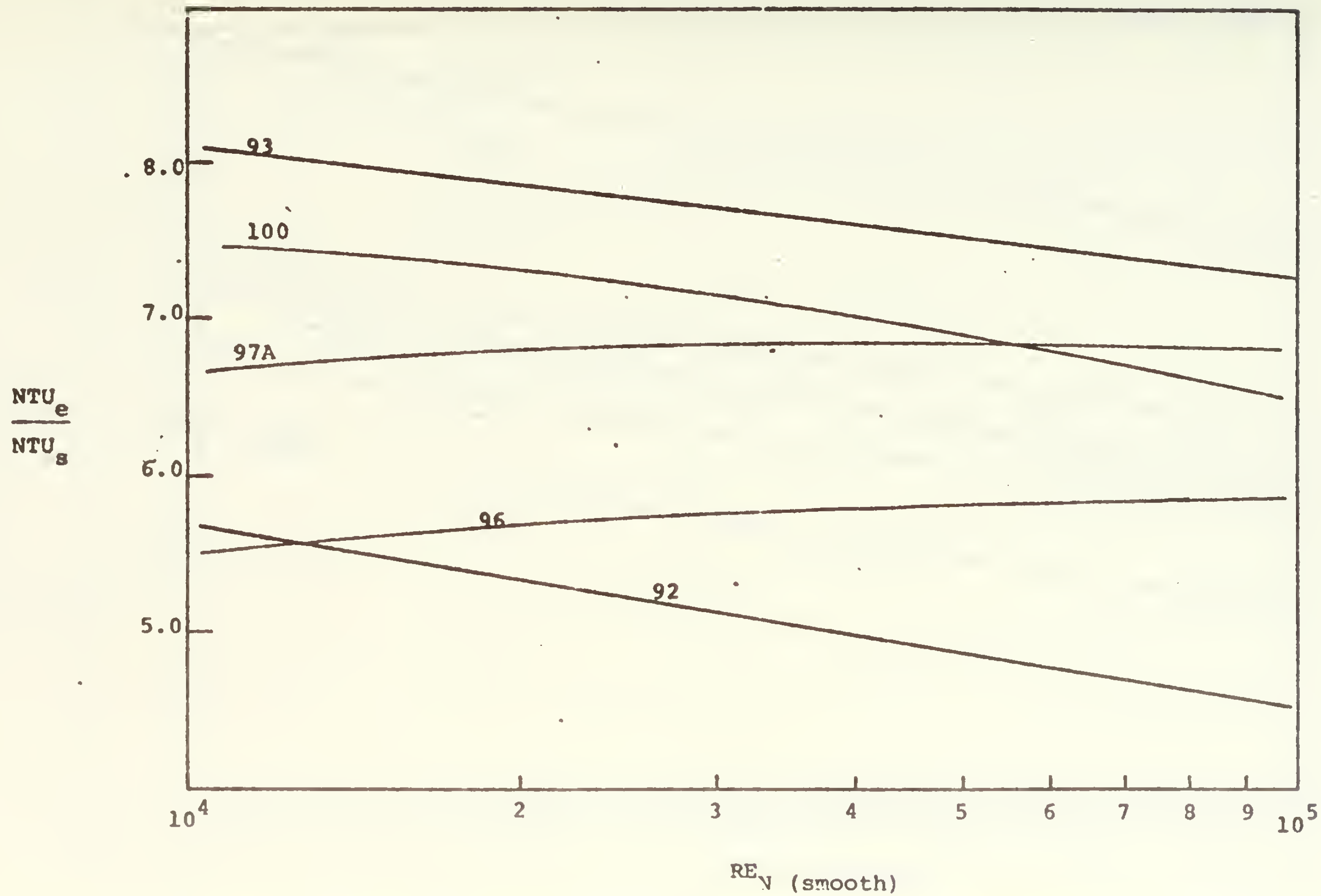
<u>Figure 13</u>: Shows the ratio of ordinates [NTU_e/NTU_s] verses RE_N for <u>Case A</u>. [Same heat exchanger shape and volume] Since both the smooth and enhanced surfaces have identical nominal diameters, the nominal Reynolds number is also identical for each surface. [see equation 27; results read from Appendix II]

<u>Figure 14</u>: Shows the ratio of abscissas [P_e/P_s] verses RE_N for <u>Case A</u>. Since both the smooth and enhanced surfaces have identical nominal diameters, the nominal Reynolds number is also identical for each surface. [see equation 27;results read from Appendix II]

<u>Figure 15</u>: Shows the ratio of ordinates [NTU_e/NTU_s] verses RE_N for <u>Case B</u>. [P=constant and V=constant]

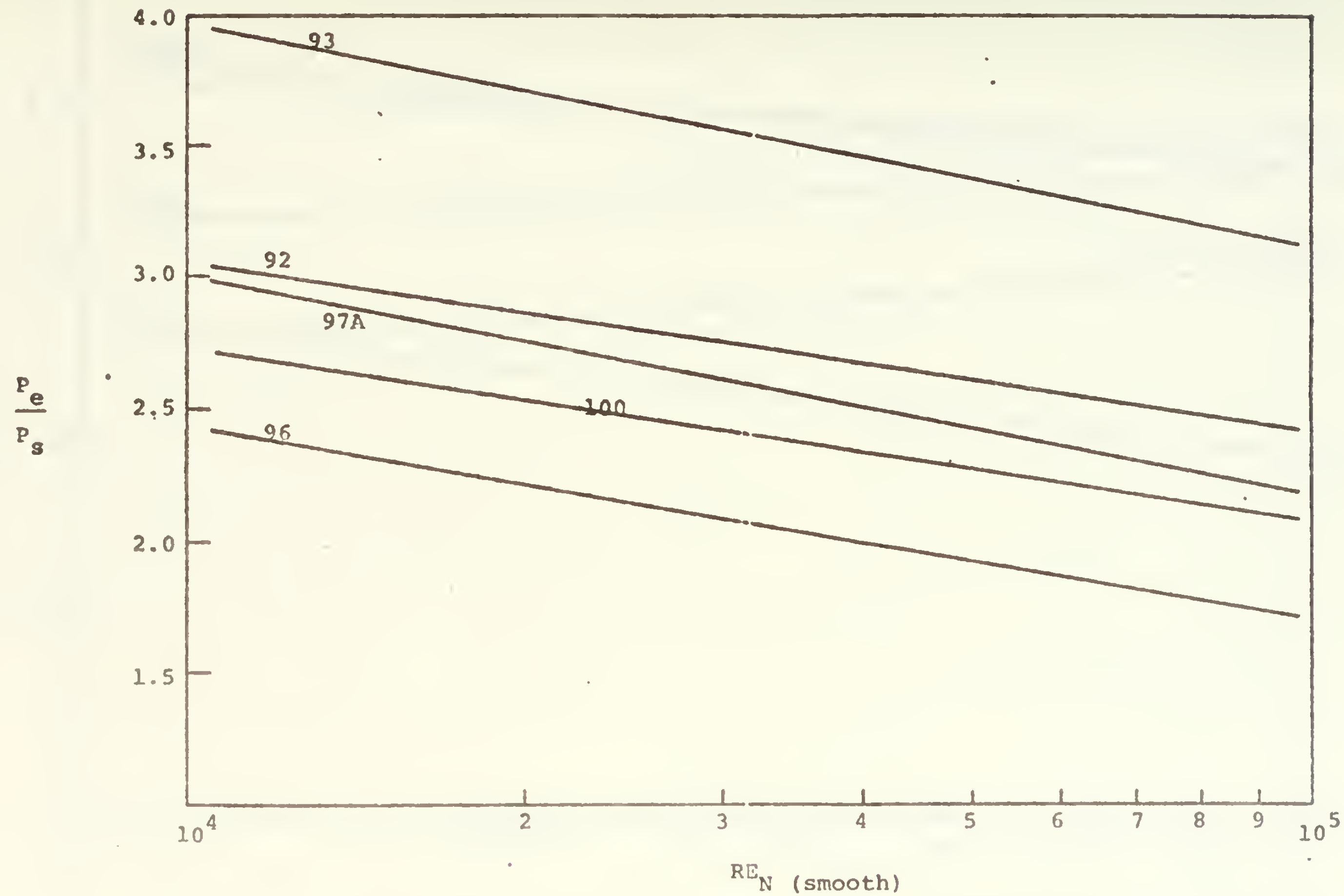
<u>Figure 16</u>: Shows the ratio of abscissas <u>or</u> ordinates to produce [V_s/V_e] verses RE_N. Recall that each axis of Figure 10 is <u>inversely</u> proportional to volume. This is <u>Case C.</u>

FIGURE 13

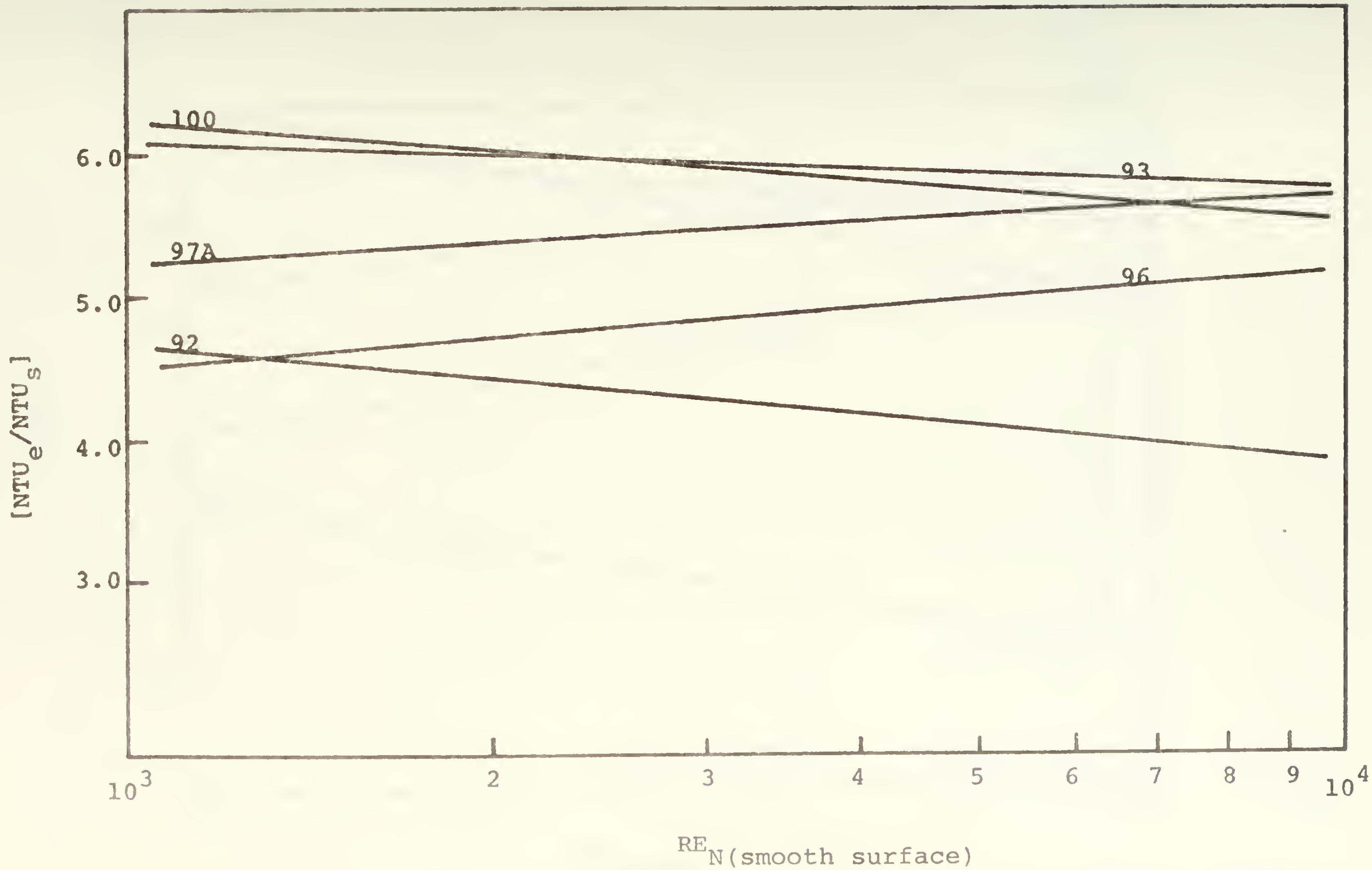


Performance Comparison Results (NTU_e/NTU_s) for Case A. Both the surfaces have $D_N \approx 1.60$ inches and, therefore, RE_N, is constant.

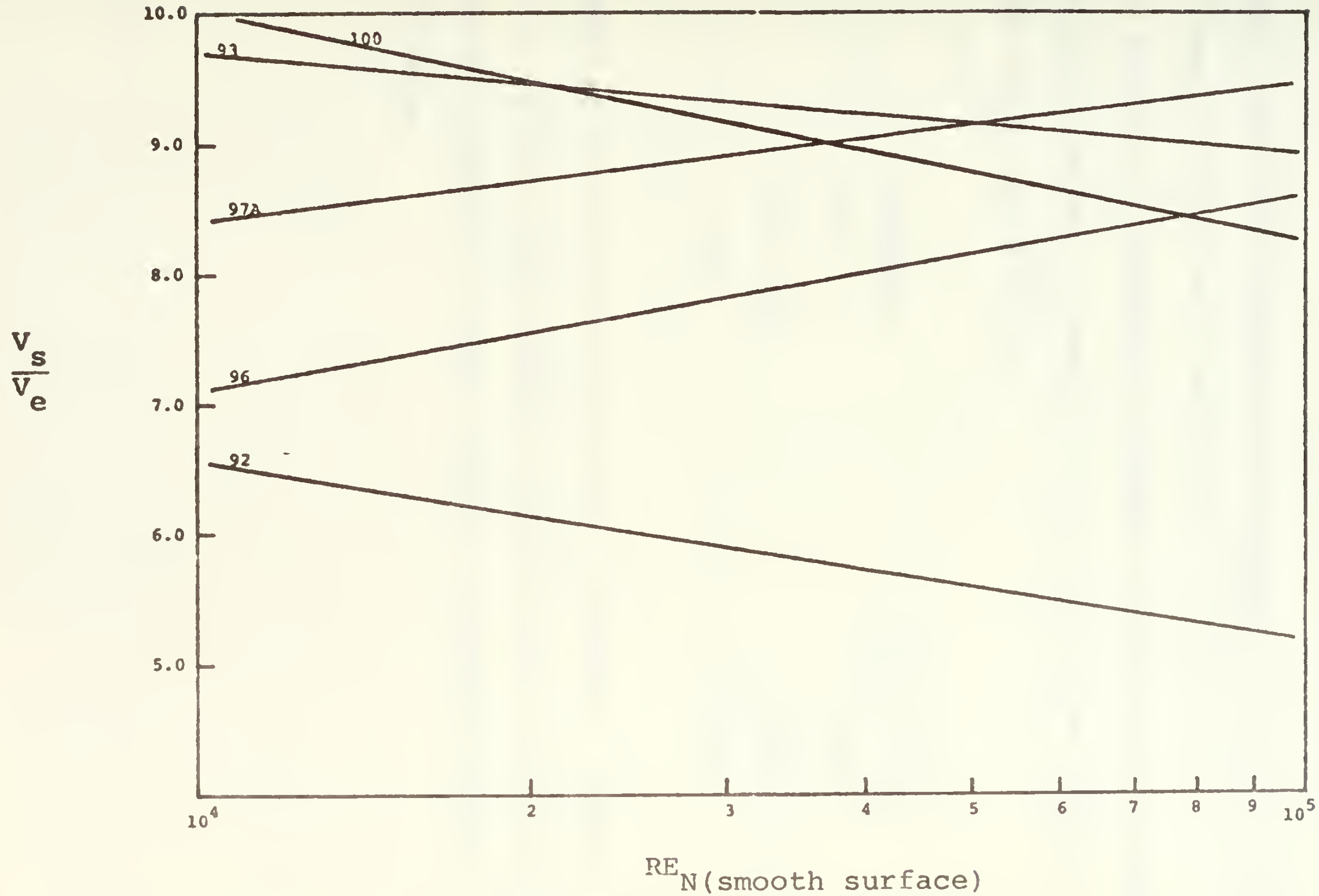
FIGURE 14



Performance comparison Results (P_e/P_s) for Case A. Both the enhanced and smooth surface have D_N≈1.60 inches and, therefore, $\overline{RE_N}$, is constant.



<u>FIGURE 15</u>: Performance Comparison Results for <u>CASE B</u> [i.e. V = Constant and P = constant], $D_N \simeq 1.60$ inches for both enhanced and smooth surfaces.



<u>FIGURE 16</u>: Performance comparison results for <u>CASE C</u> [i.e. NTU = constant, P = constant] $D_N \simeq 1.60$ inches for both enhanced & plain surface.

Figure 17: Shows the ratio of abscissas $[P_s/P_e]$ verses RE_N for Case D. [NTU=constant, and V=constant]

Higher ratios are preferred for Figures 13,15,16, and 17 while a lower ratio is most desirable in Figure 14.

As a matter of secondary interest, of the five surfaces with identical nominal diameters, there are three discrete values of the ratio $[A_F\ b]/V_{os}$:

Surface	$[A_F\ b]/V_{os}$
92 & 93	1.4282
96 & 97A	1.1859
100	1.4014

Comparison of many surfaces having identical nominal diameters to a single common smooth surface having the same nominal diameter permits the best relative comparison between enhanced surfaces.

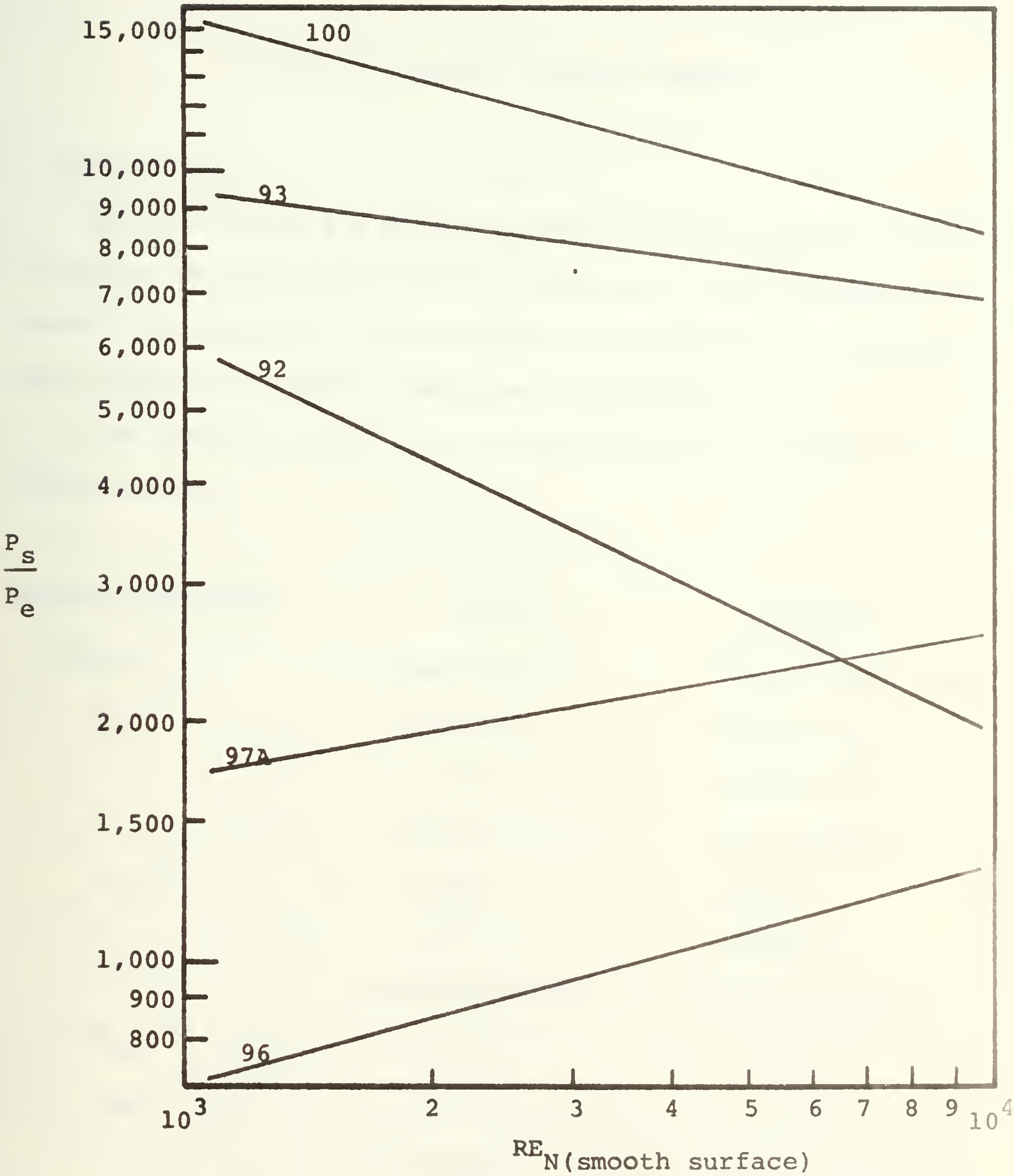


FIGURE 17: Performance comparison results for CASE D [i.e. NTU and V are constant] $D_N \simeq 1.60$ inches for both the enhanced and plain surfaces.

IV. HEAT EXCHANGER DESIGN PROBLEM

A. Description

Appendix III is a procedure which can be used for sizing crossflow finned tubular heat exchangers. The following data, taken from reference [2], is used to determine the required heat exchanger size for the given conditions.

The design is that of an intercooler for a 5000 SHP gas turbine plant.

DESIGN CONSTRAINT	SIDE I	SIDE II
Surface	Smooth Tube	Finned Tube [98C]
D_h	0.0625 FT	0.0445 FT
δ	N/A	0.0010 FT
β	5.9291 FT^2/FT^3	61.9 FT^2/FT^3
A_δ/A_T	N/A	0.8350
ℓ	N/A	0.3445 IN
W	400,000 LB/HR	200,667 LB/HR
T_{in}	60° F	260° F
T_{out}	82° F	80° F
ΔP	1.554 PSI	0.3060 PSI
μ	2.360 LB/HR-FT	0.0482 LB/HR-FT
C_p	1.000 BTU/LB-°F	0.2436 BTU/LB-°F
ρ_m	62.38 LB/FT^3	0.1755 LB/FT^3

DESIGN CONSTRAINT	SIDE I	SIDE II
P_r	6.8000	0.7000
K_m	N/A	222 BTU/HR-FT-°F
F	2.725 IN	N/A
b	1.750 IN	N/A
A_c	0.4418 IN^2	Calculate

Using the design procedure outlined in Appendix III, the principle dimensions are:

X = 6.43 FT

Y = 26.11 FT

Z = 0.63 FT

Total Volume = 105.46 FT^3

This is a possible heat exchanger design that satifies the given constraints.

B. Attempt to Minimize Total Volume

If minimization of total volume is a required design constraint, the engineer has several options the limits of which might be bounded by:

(A) Design a heat exchanger to the given specifications using each known surface for which f and j data are available. In this study sixteen surfaces are considered.

(B) Noting the marked superiority of reference [2] figure

(B) continued

94, apply the method developed in Chapter II to estimate total volume savings relative to the base or reference heat exchanger designed in part I.

Option (A) will be eliminated as being impractical due to the required effort.

Option (B) will be shown to be the most direct and highly accurate approach.

The f and j verses RE data for reference [2] figure 98C and 94 are presented in Figure 18. The problem presented corresponds to Case C described in Figure 16.

Using the heat exchanger design procedure outlined in Appendix III and replacing the finned tube side with the surface of figure 94 [Ref 2]:

$$X = 1.50 \text{ FT}$$

$$Y = 10.89 \text{ FT}$$

$$Z = 1.76 \text{ FT}$$

$$\text{Total Volume} = 28.68 \text{ FT}^3$$

A 73% reduction in volume would result by employing the surface of figure 94 [Ref 2].

In order to use the comparison method developed in Chapter II to quantitatively predict the volume savings, a performance parameter plot is constructed for each surface considered. Figure 19 presents the only two surfaces being considered in this example.

<u>FIGURE 18</u>: f and j verses RE data for the two surfaces considered in the design example.

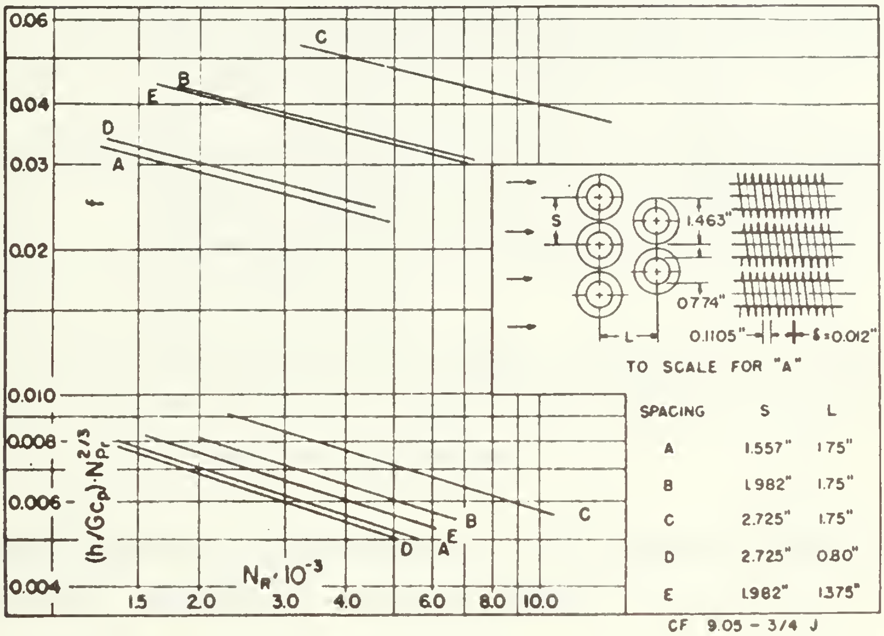


FIG. 98

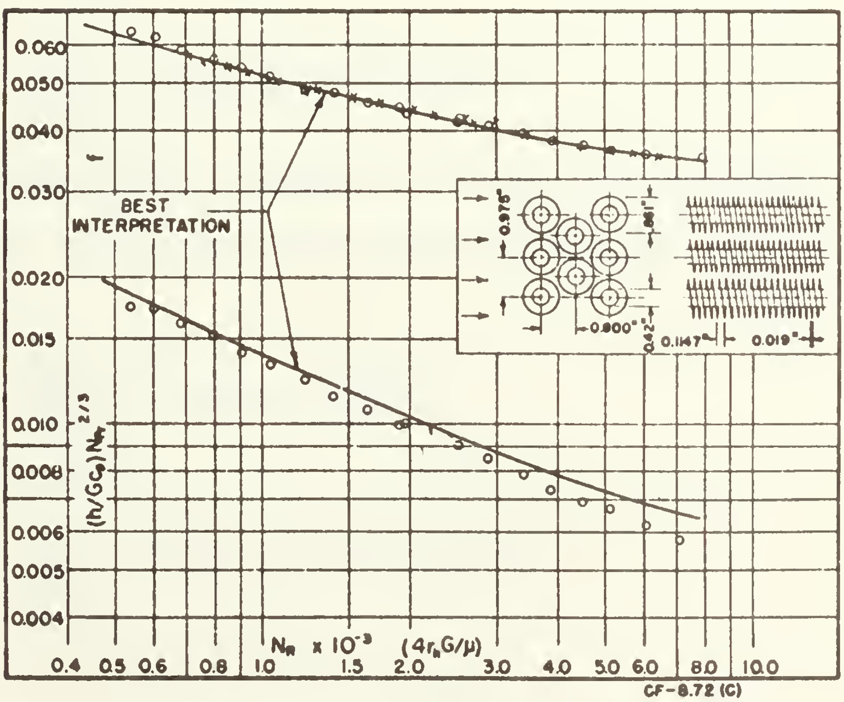


FIG. 94

As outlined in Chapter II for Case C:

(1) From the design point of the reference heat exchanger (Figure 98C) graphically or analytically construct a line of slope 1.

(2) The intersection of this line with the new surface (Figure 94) will yield the performance parameters at the design point of the new heat exchanger.

(3) The predicted volume reduction is found from the ratio of either the ordinates or abscissas of the two surfaces.

The performance parameters of the reference heat exchanger design are: [Reference 2 figure 98C]

$$\frac{NTU}{V_{os}} = 401.20 \ [1/IN^2]$$

$$\frac{P}{V_{os}} = 2.93 \times 10^{12} \ [1/IN^4]$$

The performance parameters of the heat exchanger that would result from using the surface described by reference[2] figure 94:

$$\frac{NTU}{V_{os}} = 1634.68 \ [1/IN^2]$$

$$\frac{P}{V_{os}} = 1.19 \times 10^{13} \ [1/IN^4]$$

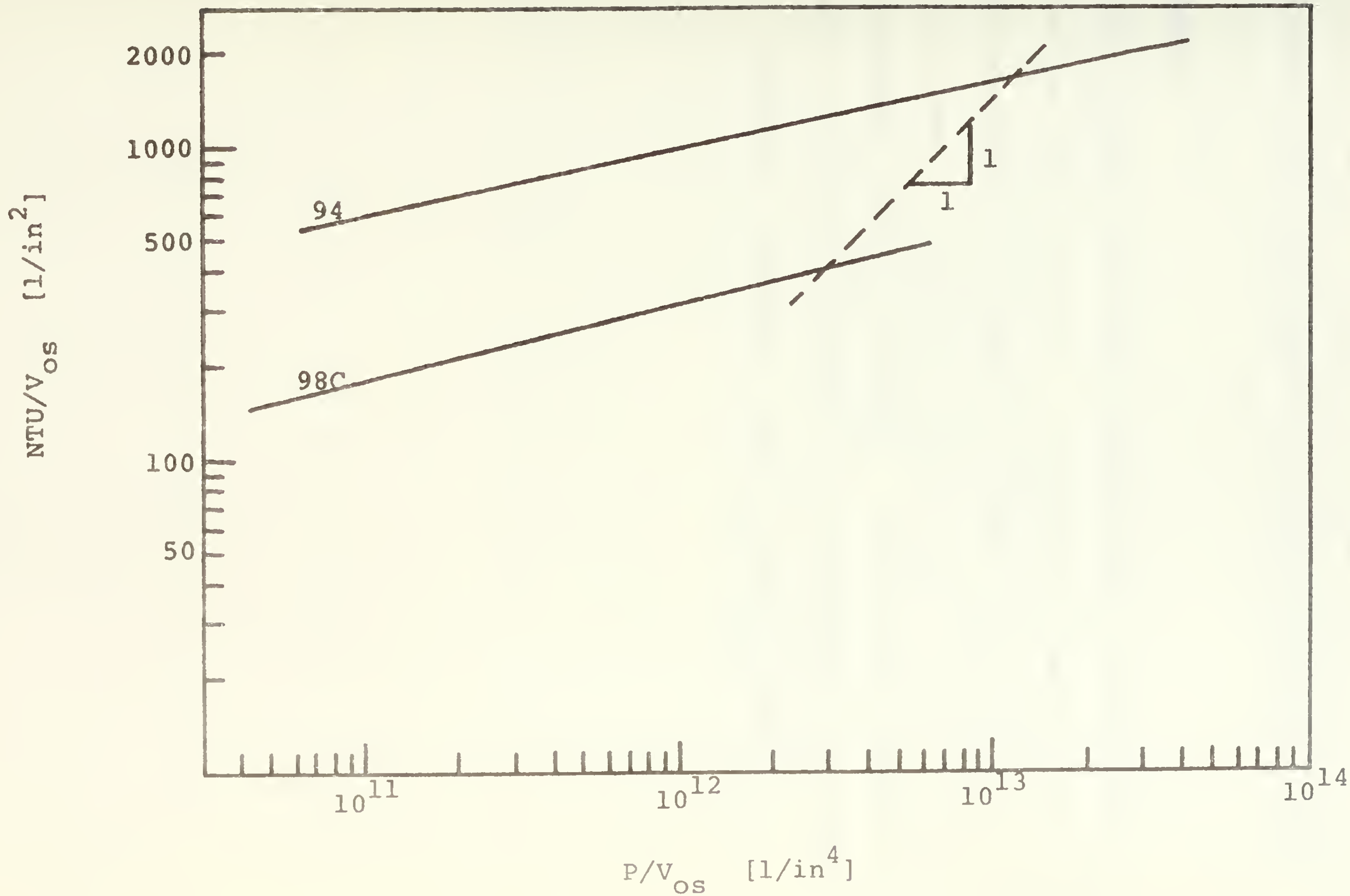


FIGURE 19: Performance Parameter Plot for the two surfaces used in the Design Example

In other words, a predicted volume reduction of 75% is anticipated using the comparison method developed herein.

C. Summary

Two surfaces have been compared from a minimization of total volume point-of-view. It has been shown that the performance parameters developed in Chapter II allow us to compare any number of surfaces without going through a complete heat exchanger design calculation. Where there may well be hundreds of different surfaces to be considered, the value of such a comparison technique cannot be overestimated.

V. CONCLUSIONS

1. The comparison method of Soland [5] was used as a basis in developing a comparison method for finned tube heat exchanger surfaces.

2. Heat Exchanger Performance can be compared on four different bases:

   a. Same shape and volume of heat exchanger

   b. Same exchanger volume and pumping power

   c. Same pumping power and NTU

   d. Same volume and NTU

3. All of Kays and London [2] finned tube surfaces and several surfaces supplied by the Trane Corporation were compared on the above basis.

4. The method developed herein was applied to a practical heat exchanger design problem and found to accurately predict relative volume savings between different surfaces.

REFERENCES

1. Rohsenow, W.M. and Choi, H., Heat, Mass, and Momentum Transfer, Prentice-Hall, Inc., Englewood Cliff, New Jersey, 1961.

2. Kays, W.M. and London, A.L., Compact Heat Exchangers, McGraw-Hill Book Company, Inc., New York, 1958.

3. Kern, D.Q. and Kraus, A.D., Extended Surface Heat Transfer, McGraw-Hill Book Company, Inc., New York, 1972.

4. Rohsenow, W.M. and Hartnett, J.P., Handbook of Heat Transfer, McGraw-Hill Book Company, Inc., New York, 1973.

5. Soland, J.G., "Performance Ranking of Plate Finned Heat Exchanger Surfaces", M.I.T. Ocean Engineering Thesis, May, 1975.

6. Sheldon, E.S., "Heat Exchanger Performance Ranking", M.I.T. Ocean Engineering Thesis, May, 1976.

7. Zhukaukas, A., "Heat Transfer From Tubes in Crossflow", Advances in Heat Transfer, Vol 8, pp 93-160, 1972.

8. Personal communication, Director Research and Development, Harrison Radiator Divison, General Motors Corporation; with W.M. Rohsenow.

9. Mack, W.M., "Evaluation of the Heat Transfer Performance of Three Enhanced Surfaces Including and Enhanced Surface Heat Exchanger Performance Comparison", Ph.D disertation M.I.T., 1974.

REFERENCES (continued)

10. Fraas and Ozisik, Heat Exchanger Design, John Wiley and Sons, Inc., New York, 1965.

APPENDIX I

A. Purpose

A frequent criticism of academic solutions to "optimization" of design is: "So what, . . . how does this help the engineer in industry?" [8]

It is hoped that this section will give the "engineer in industry" the needed background and technique to effectively compare finned tubular heat exchanger surfaces.

A step-by-step flow diagram using the performance comparison equations derived in the text of the thesis is presented in Figure A-1. This procedure is easily programmed for use on either a (1) standard computer or (2) programmable calculator.

In keeping with the spirit of the criticism of Reference [8], a program for use on a personal programmable calculator will be presented herein. It is intended that this simplified, easy-to-use approach will demonstrate that the "engineer in industry" needn't resort to often costly computer time/programs in order to effectively "optimize" a design.

B. Proposed Program(s) for Calculating Surface Performance Parameters on a Programmable* Calculator

The following programs follow the general outline of Figure A-1:

Figure A-II: For calculating Surface Performance
Parameters for the case in which the

CALCULATION OF FINNED TUBULAR HEAT EXCHANGER SURFACE PERFORMANCE PARAMETERS

INPUT

Linearized equations of the form $A[RE]^{B}$ for f and j for both the enhanced and "if smooth" surface:

$A_{f(e)}$; $B_{f(e)}$; $A_{j(e)}$; $B_{j(e)}$

$A_{f(s)}$; $B_{f(s)}$; $A_{j(s)}$; $B_{j(s)}$

Geometrical Factors: X_L; X_T; D_0; D_{2h}; δ; ρ; ℓ; β; $[A_{fin}/A_{Tot}]$;

Other: Air properties, fin material and shape, Reynolds Number range

DOES THE MINIMUM FREE FLOW AREA OCCUR IN A DIAGONAL? [Is $2X_d < X_T$?]

NO

Calculate the Nominal Diameter:

$$D_{N_t} = \frac{4D_0X_L[X_T-1]}{\pi}$$

$$\frac{[f_N]}{[f]_t} = \frac{16[X_T-1]^2 D_N}{\beta^2 X_T^2 D_{2h}^3}$$

$$\frac{A_F b}{V_{os}} = \frac{2[X_T-1]X_L}{X_T X_L - 0.25\pi}$$

YES

Calculate the Nominal Diameter:

$$D_{N_d} = \frac{8D_oX_L[X_D-1]}{\pi}$$

$$\frac{[f_N]}{[f]_d} = \frac{64[X_D-1]^2 D_N}{\beta^2 X_T^2 D_{2h}^3}$$

$$\frac{A_F b}{V_{os}} = \frac{4[X_D-1]X_L}{X_D^2\sin\alpha - 0.25\pi}$$

Compute the ratio of friction factor based on base Area $[A_b]$ to the friction factor based on total area $[A_T]$

Compute the ratio of Flow Area $[A_F]$ times equivalent flow length [b] to the Volume on one side $[V_{os}]$ of a heat exchanger

$$\frac{RE_N}{RE} = \frac{\beta X_T X_L D_o}{\pi}$$

Compute ratio of Reynolds Number based on minimum free flow are $[A_c]$ to Reynolds Number based on free flow area $[A_F]$

A

$$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3 A_F b}{D_N^4 V_{os}}$$

Compute Power per unit Volume on one side of a heat exchanger using the specified ENHANCED surface.
Note: $f_{(e)} = A_{f(e)} RE^{B_{f(e)}}$

$$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$$

Compute Power per unit Volume on one side of a heat exchanger using the specified surface as "if SMOOTH"

Note: $f_{(s)} = A_{f(s)} RE^{B_{f(s)}}$
$RE = RE_N$ for smooth surface

FIGURE A-1: Performance Parameters

Compute fin efficiency assuming a Longitudinal fin of Rectangular Profile:

$$\eta_{f-L} = \frac{\tanh m\ell}{m\ell}$$

The fin material for a given surface is known. Assume a gas and a operating temperature in order to calculate fin efficiency for comparison purposes.

What geometric shape describes the fin being employed on the tube surface?

Radial Fin of Rectangular Profile

$$\Delta\eta_{correction} = C_\delta - D_\delta \ln(\eta_{f-L})$$

where:

$$C_\delta = -.00259 - .0084 \ln \frac{r_o}{r_e}$$

$$D_\delta = -.02436 - .4521 \ln \frac{r_o}{r_e}$$

(Ref 3 page 264)

Tapered (hyperbolic) Radial fin

$$\Delta\eta_{correction} = 0$$

(Ref 4 page 3-116)

Rectangular Profile fin sheet

$\Delta\eta_{correction} =$

Same as Radial Fin of Rectangular Profile using r_e' (staggered bank)

$$r_e' = \left[\frac{2\sqrt{3}}{\pi}\right]^{.50} \frac{X_d D_o}{2}$$

$$\ell = r_e' - r_o$$

(Ref 4 page 3-116)

$$\eta_{f\text{-corrected}} = \eta_{f-L} - \Delta\eta_{correction}$$

Compute Surface η:

$$\eta_o = 1 - \frac{A_{fin}}{A_T}[1 - \eta_{f-cr}]$$

$$\frac{NTU_e}{V_{os}} = \frac{j_N \, RE_N \, A_F \, b}{D_N^2 \, V_{os}}$$

Compute number of NTU per unit Volume on one side of a heat exchanger using the specified ENHANCED surface.

Note: $j_N = (j\eta_o D_N)/D_{2h}]$ and

$$j_{(e)} = A_{j(e)} RE^{B_{j(e)}}$$

$$\frac{NTU_S}{V_{os}} = \frac{j_{(s)} \, RE \, A_F \, b}{D_N^2 \, V_{os}}$$

Compute number of NTU per unit Volume on one side of a heat exchanger using the specified surface as "if SMOOTH".

Note: $j_{(s)} = A_{j(s)} RE^{B_{j(s)}}$

Add selected increment to RE. Test to see if range of data is exceeded?

NO → Go to A

YES → END

	minimum free flow area occurs transversely to the flow direction.
<u>Figure A-II-A</u>:	Definition of geometric constraints/ constants to be stored in memory.
<u>Figure A-II-B</u>:	Sample output for transverse case.
<u>Figure A-III</u>:	For calculating Surface Performance Parameters for the case in which the minimum free flow area occurs in the diagonals. [see Figure 4]
<u>Figure A-III-A</u>:	Definition of geometric constraints/ constants to be stored in memory.
<u>Figure A-III-B</u>:	Sample output for diagonal case.

C. <u>Operation</u>

<u>Step</u>	<u>Description</u>
1.	Select and Load Program
2.	Store Constants as outlined in either Figure A-II-A or A-III-A.
3.	Press Key <u>A</u>
4.	Compile Output
5.	Stop Program when range of RE data is exceeded.

PROGRAM* FOR CALCULATING SURFACE PERFORMANCE PARAMETERS FOR A CROSSFLOW FINNED TUBULAR HEAT EXCHANGER IN WHICH THE MINIMUM FREE FLOW AREA OCCURS TRANSVERSLY TO THE FLOW DIRECTION

FIGURE A-II

001	*LBLA	21 11	057	2	02	113	×	-35	169	P⇄S	16-51
002	RCLA	36 11	058	X⇄Y	-41	114	PRTX	-14	170	RCL8	36 08
003	PRTX	-14	059	÷	-24	115	RCL0	36 00	171	CHS	-22
004	SPC	16-11	060	RCLE	36 15	116	RCLC	36 13	172	×	-35
005	RCLD	36 14	061	1	01	117	×	-35	173	RCL6	36 05
006	RCLC	36 13	062	-	-45	118	RCL4	36 04	174	+	-55
007	×	-35	063	×	-35	119	3	03	175	P⇄S	16-51
008	4	04	064	RCLD	36 14	120	+	-55	176	RCL8	36 08
009	×	-35	065	×	-35	121	Y^X	31	177	×	-35
010	Pi	16-24	066	STOE	35 15	122	RCLA	36 11	178	+	-55
011	÷	-24	067	P⇄S	16-51	123	4	04	179	PRTX	-14
012	RCLE	36 15	068	RCL9	36 09	124	Y^X	31	180	RCL0	36 00
013	1	01	069	LN	32	125	÷	-24	181	×	-35
014	-	-45	070	P⇄S	16-51	126	RCL3	36 03	182	RCL0	36 00
015	×	-35	071	RCL9	36 09	127	×	-35	183	-	-45
016	STOA	35 11	072	×	-35	128	RCLE	36 15	184	1	01
017	RCLD	36 14	073	RCL8	36 08	129	×	-35	185	+	-55
018	RCLE	36 15	074	+	-55	130	PRTX	-14	186	RCL3	36 03
019	×	-35	075	STO8	35 08	131	RCL0	36 00	187	×	-35
020	RCLB	36 12	076	P⇄S	16-51	132	P⇄S	16-51	188	P⇄S	16-51
021	×	-35	077	RCL9	36 09	133	RCL4	36 04	189	RCL5	36 05
022	RCLC	36 13	078	LN	32	134	1	01	190	÷	-24
023	×	-35	079	P⇄S	16-51	135	+	-55	191	RCLA	36 11
024	Pi	16-24	080	RCL7	36 07	136	2	02	192	÷	-24
025	÷	-24	081	×	-35	137	÷	-24	193	RCLC	36 13
026	STOC	35 13	082	RCL6	36 06	138	Y^X	31	194	×	-35
027	RCLI	36 46	083	+	-55	139	RCL3	36 03	195	RCL0	36 00
028	X⇄Y	-41	084	STO6	35 06	140	2	02	196	1	01
029	÷	-24	085	PREG	16-13	141	×	-35	197	P⇄S	16-51
030	STOI	35 46	086	P⇄S	16-51	142	RCL2	36 02	198	RCL4	36 04
031	1	01	087	PREG	16-13	143	×	-35	199	+	-55
032	6	06	088	P⇄S	16-51	144	RCL5	36 05	200	Y^X	31
033	RCLB	36 12	089	*LBLC	21 13	145	÷	-24	201	×	-35
034	RCLE	36 15	090	RCLI	36 46	146	P⇄S	16-51	202	RCLE	36 15
035	×	-35	091	RCL0	36 00	147	RCL5	36 05	203	×	-35
036	X^2	53	092	+	-55	148	÷	-24	204	PRTX	-14
037	÷	-24	093	STO0	35 00	149	P⇄S	16-51	205	P⇄S	16-51
038	RCL5	36 05	094	PRTX	-14	150	$\sqrt{X}$	54	206	RCL0	36 00
039	3	03	095	RCL2	36 02	151	×	-35	207	RCLC	36 13
040	Y^X	31	096	Y^X	31	152	RCL1	36 01	208	×	-35
041	÷	-24	097	RCL1	36 01	153	×	-35	209	P⇄S	16-51
042	RCLE	36 15	098	×	-35	154	STOD	35 14	210	RCL7	36 07
043	1	01	099	RCLB	36 12	155	e^X	33	211	1	01
044	-	-45	100	×	-35	156	RCLD	36 14	212	+	-55
045	X^2	53	101	RCLA	36 11	157	CHS	-22	213	Y^X	31
046	×	-35	102	4	04	158	e^X	33	214	RCL6	36 06
047	RCLA	36 11	103	Y^X	31	159	-	-45	215	×	-35
048		-35	104	÷	-24	160	LSTX	16-63	216	RCLA	36 11
049	STOB	35 12	105	RCL0	36 00	161	RCLD	36 14	217	X^2	53
050	RCLE	36 15	106	RCLC	36 13	162	e^X	33	218	÷	-24
051	RCLD	36 14	107	×	-35	163	+	-55	219	RCLE	36 15
052	×	-35	108	PRTX	-14	164	÷	-24	220	×	-35
053	Pi	16-24	109	3	03	165	RCLD	36 14	221	PRTX	-14
054	4	04	110	Y^X	31	166	÷	-24	222	SPC	16-11
055	÷	-24	111	×	-35	167	ENT↑	-21	223	P⇄S	16-51
056	-	-45	112	RCLE	36 15	168	LN	32	224	STOC	35 13

*For Hewlett-Packard 67/97 Programmable Calculator

FIGURE A-II-A

GEOMETRIC CONSTRAINTS/CONSTANTS TO BE STORED IN MEMORY

[Transverse Case]

REG	QTY	DESCRIPTION	REG	QTY	DESCRIPTION
0	RE	Leave Blank. This register is used in the program to store the value of RE being used in each iteration.	10	A_f/A_T	Ratio of fin area to total area.
1	$A_{f(e)}$	From linearized equation for friction factor (f) of the <u>enhanced</u> surface of the form: $A\ RE^{B}$	11	ℓ	Fin length [inches]
2	$B_{f(e)}$	From linearized equation for friction factor (f) of the <u>enhanced</u> surface of the form: $A\ RE^{B}$	12	Ψ	Property constant defined by: $[C_p\ \mu]/\ [K_m\ P_r^{2/3}]$ [ND]
3	$A_{f(s)}$	From linearized equation for friction factor (f) of the <u>if smooth</u> surface of the form: $A\ RE^{B}$	13	$A_{j(e)}$	From linearized equation for Colburn modulus (j) of the <u>enhanced</u> surface of the form: $A\ RE^{B}$
4	$B_{f(s)}$	From linearized equation for friction factor (f) of the <u>if smooth</u> surface of the form: $A\ RE^{B}$	14	$B_{j(e)}$	From linearized equation for Colburn modulus (j) of the <u>enhanced</u> surface of the form: $A\ RE^{B}$
5	D_{2h}	Specified hydraulic diameter of the flow passage [inches]	15	δ	Specified fin thickness <u>or</u> average thickness, if tapered. [inches]
6	A_1	Constants derived from graph on page 244 of Reference [3]. They are used in the computation of a correction factor applied to η_{f-L} in order to permit the use of the simplier $\frac{\tanh m\ell}{m\ell}$ equation for fin temperature effectiveness in lieu of Bessel function type equations.	16	$A_{j(s)}$	From linearized equation for Colburn modulus (j) of the "<u>if smooth</u>" surface of the form: $A\ RE^{B}$
7	A_2		17	$B_{j(s)}$	From linearized equation for Colburn Modulus (j) of the "<u>if smooth</u>" surface of the form: $A\ RE^{B}$
8	A_3		18	Fin ID	Input one: 0.000 if fin is tapered; -1.000 if fin is a radial fin of rectangular profile
9	A_4		19	ρ	Ratio of outside tube radius to outside fin radius.
A	Data ID	Any desired number to permit the user to identify the surface being processed.	D	X_L	Longitudinal tube pitch. Ratio of longitudinal tube spacing to tube outside diameter.
B	β	Specified heat transfer area per total volume ratio. [in^2/in^3]	E	X_T	Transverse tube pitch. Ratio of transverse tube spacing to tube outside diameter.
C	D_o	Specified tube outside diameter. [inches]	I	Increment	Desired increment of RE_N. A value of 10,000 is recommended.

FIGURE A-II-B

SAMPLE OUTPUT

[Transverse Case]

FIGURE ID# → 100.0000

	0.0000	0	Print out of Registers 0-9 and A-I
	0.1223	1	
	-0.2073	2	
	0.3514	3	
	-0.0896	4	
	0.1430	5	
	0.0054	6	
	-0.0084	7	
	0.4098	8	
	-0.4521	9	
	1.6402	A	D_N
f_N/f	14.4115	B	
	10.2285	C	RE_N/RE
	2.1542	D	
$[A_F\ b]/V_{os}$	1.4014	E	
	977.6564	I	

	0.8390	0	Print out of Registers 10-19 and A-I
	0.3240	1	
	0.0001	2	
	0.1650	3	
	-0.3986	4	
	0.0130	5	
	0.5830	6	
	-0.3960	7	
	-1.0000	8	
	0.3829	9	

	977.6564	[Value of RE being processed
Value of RE_N being processed]	10000.0000	
	8.186009734+10	P_e/V_{os}
P_s/V_{os}	2.981639001+10	
	0.9328	η_δ-corrected
NTU_e/V_{os}	598.0269	
	79.1432	NTU_s/V_{os}

FIGURE A-III

PROGRAM* FOR CALCULATING SURFACE PERFORMANCE PARAMETERS FOR A CROSSFLOW FINNED TUBULAR HEAT EXCHANGER IN WHICH THE MINIMUM FREE FLOW AREA OCCURS IN THE DIAGONALS

001	*LBLA	21 11	057	P⇄S	16-51	113	3	03	169	ENT↑	-21
002	RCLA	36 11	058	RCL5	36 05	114	Yˣ	31	170	LN	32
003	PRTX	-14	059	3	03	115	×	-35	171	P⇄S	16-51
004	RCL0	36 00	060	Yˣ	31	116	RCLE	36 15	172	RCL8	36 08
005	X²	53	061	÷	-24	117	×	-35	173	×	-35
006	RCLE	36 15	062	RCL0	36 00	118	PRTX	-14	174	RCL6	36 06
007	SIN	41	063	1	01	119	RCL0	36 00	175	+	-55
008	×	-35	064	-	-45	120	RCLC	36 13	176	P⇄S	16-51
009	Pi	16-24	065	X²	53	121	×	-35	177	RCL9	36 09
010	4	04	066	×	-35	122	RCL4	36 04	178	×	-35
011	÷	-24	067	RCLA	36 11	123	3	03	179	+	-55
012	-	-45	068	×	-35	124	+	-55	180	RCL0	36 00
013	4	04	069	STOB	35 12	125	Yˣ	31	181	×	-35
014	X⇄Y	-41	070	P⇄S	16-51	126	RCLA	36 11	182	RCL0	36 00
015	÷	-24	071	RCL9	36 09	127	4	04	183	-	-45
016	RCL0	36 00	072	LN	32	128	Yˣ	31	184	1	01
017	1	01	073	P⇄S	16-51	129	÷	-24	185	+	-55
018	-	-45	074	RCL9	36 09	130	RCL3	36 03	186	RCL3	36 03
019	×	-35	075	×	-35	131	×	-35	187	×	-35
020	RCLD	36 14	076	RCL8	36 08	132	RCLE	36 15	188	P⇄S	16-51
021	×	-35	077	+	-55	133	×	-35	189	RCL5	36 05
022	STOE	35 15	078	STO8	35 08	134	PRTX	-14	190	÷	-24
023	RCLD	36 14	079	P⇄S	16-51	135	RCL0	36 00	191	RCLA	36 11
024	RCLC	36 13	080	RCL9	36 09	136	P⇄S	16-51	192	÷	-24
025	×	-35	081	LN	32	137	RCL1	36 01	193	RCLC	36 13
026	8	08	082	P⇄S	16-51	138	1	01	194	×	-35
027	×	-35	083	RCL7	36 07	139	+	-55	195	RCL0	36 00
028	Pi	16-24	084		-35	140	2	02	196	1	01
029	÷	-24	085	RCL6	36 06	141	÷	-24	197	P⇄S	16-51
030	RCL0	36 00	086	+	-55	142	Yˣ	31	198	RCL4	36 04
031	1	01	087	STO6	35 06	143	RCL3	36 03	199	+	-55
032	-	-45	088	PREG	16-13	144	2	02	200	Yˣ	31
033	×	-35	089	P⇄S	16-51	145		-35	201	×	-35
034	STOA	35 11	090	PREG	16-13	146	RCL2	36 02	202	RCLE	36 15
035	RCLD	36 14	091	P⇄S	16-51	147		-35	203	×	-35
036	RCLC	36 13	092	0	00	148	P⇄S	16-51	204	PRTX	-14
037	×	-35	093	STO0	35 00	149	RCL5	36 05	205	P⇄S	16-51
038	RCLB	36 12	094	*LBLC	21 13	150	÷	-24	206	RCL0	36 00
039	×	-35	095	RCLI	36 46	151	P⇄S	16-51	207	RCLC	36 13
040	Pi	16-24	096	RCL0	36 00	152	√X	54	208	×	-35
041	÷	-24	097	+	-55	153		-35	209	P⇄S	16-51
042	P⇄S	16-51	098	STO0	35 00	154	RCL1	36 01	210	RCL7	36 07
043	RCL5	36 05	099	RCL2	36 02	155	×	-35	211	1	01
044	×	-35	100	Yˣ	31	156	STOD	35 14	212	+	-55
045	STOC	35 13	101	RCL1	36 01	157	eˣ	33	213	Yˣ	31
046	RCLI	36 46	102	×	-35	158	RCLD	36 14	214	RCL6	36 06
047	X⇄Y	-41	103	RCLB	36 12	159	CHS	-22	215	×	-35
048	÷	-24	104	×	-35	160	eˣ	33	216	RCLA	36 11
049	STOI	35 46	105	RCLA	36 11	161	-	-45	217	X²	53
050	6	06	106	4	04	162	LSTX	16-63	218	÷	-24
051	4	04	107	Yˣ	31	163	RCLD	36 14	219	RCLE	36 15
052	RCLB	36 12	108	÷	-24	164	eˣ	33	220	×	-35
053	RCL5	36 05	109	RCL0	36 00	165	+	-55	221	PRTX	-14
054	×	-35	110	RCLC	36 13	166	÷	-24	222	SPC	16-11
055	X²	53	111	×	-35	167	RCLD	36 14	223	P⇄S	16-51
056	÷	-24	112	PRTX	-14	168	÷	-24	224	GTOC	22 13

*For Hewlett-Packard 67/97 Programmable Calculator

FIGURE A-III-A

GEOMETRIC CONSTRAINTS/CONSTANTS TO BE STORED IN MEMORY

[Diagonal Case]

REG	QTY	DESCRIPTION	REG	QTY	DESCRIPTION
0	X_D	Specified diagonal tube pitch. Ratio of diagonal tube spacing to the tube outside diameter.	10	A_f/A_T	Ratio of fin area to total area.
1	$A_{f(e)}$	From linearized equation for friction factor (f) of the <u>enhanced</u> surface of the form: $A\ RE^{B}$	11	$\ell/\sqrt{\delta}$	Ratio of fin length to fin thickness. $[1/\sqrt{inch}]$
2	$B_{f(e)}$	From linearized equation for friction factor (f) of the <u>enhanced</u> surface of the form: $A\ RE^{B}$	12	ϕ	Property constant defined by: $[C_p\ \mu]/\ [K_m\ P_r^{2/3}]$ (ND)
3	$A_{f(e)}$	From linearized equation for friction factor (f) of the <u>if smooth</u> surface of the form: $A\ RE^{B}$	13	$A_{j(e)}$	From linearized equation for Colburn modulus (j) of the <u>enhanced</u> surface of the form: $A\ RE^{B}$
4	$B_{f(e)}$	From linearized equation for friction factor (f) of the <u>if smooth</u> surface of the form: $A\ RE^{B}$	14	$B_{j(e)}$	From linearized equation for Colburn modulus (j) of the <u>enhanced</u> surface of the form: $A\ RE^{B}$
5	D_{2h}	Specified hydraulic diameter of the flow passage [inches]	15	X_T	Transverse tube pitch. Ratio of transverse tube spacing to tube outside diameter.
6	A_1	Constants derived from graph on page 264 of Reference [3]. They are used in the computation of a correction factor applied to η_{f-L} in order to permit the use of the simplier $\frac{\text{Tanh}\ m\ell}{m\ell}$ equation for fin temperature effectiveness in lieu of Bessel function type equations.	16	$A_{j(e)}$	From linearized equation for Colburn modulus (j) of the "<u>if smooth</u>" surface of the form: $A\ RE^{B}$
7	A_2		17	$B_{j(e)}$	From linearized equation for Colburn Modulus (j) of the "<u>if smooth</u>" surface of the form: $A\ RE^{B}$
8	A_3		18	Fin ID	Input one: 0.000 if fin is tapered; -1.000 if fin is a radial fin of rectangular profile
9	A_4		19	ρ	Ratio of outside tube radius to outside fin radius.
A	Data ID	Any desired number to permit the user to identify the surface being processed.	D	X_L	Longitudinal tube pitch. Ratio of longitudinal tube spacing to tube outside diameter.
B	β	Specified heat transfer area per total volume ratio. $[in^2/in^3]$	E	α	Angle defined by longitudinal centerline and diagonal centerline: [degrees]
C	D_o	Specified tube outside diameter. [inches]	I	Increment	Desired increment of RE_N. A value of 10,000 is recommneded.

FIGURE A-IIIB

SAMPLE OUTPUT

[diagonal case]

FIGURE ID# →	98.4000		Print out of Registers 0-9 and A-I
	2.0413	0	
	0.2173	1	
	-0.2590	2	
	1.1125	3	
	-0.1971	4	
	0.1904	5	
	0.0027	6	
	-0.0084	7	
	0.2636	8	
	-0.4521	9	
	2.1214	A	← D_N
f_N/f →	13.5882	B	
	10.0860	C	← RE_N/RE
	1.0336	D	
$[A_F\ b]/V_{os}$ →	1.5332	E	
	991.4751	I	

	0.8350	0	Print out of Registers 10-19
	3.1448	1	
	0.0001	2	
	0.0813	3	
	-0.3264	4	
	3.5207	5	
	0.5965	6	
	-0.3960	7	
	-1.0000	8	
	0.5290	9	

Value of RE_N being evaluated →	10000.0000	
	3.743772534+10	← P_e/Vos
P_s/V_{os} →	1.370584009+10	
	320.2228	← NTU_e/V_{os}
NTU_s/V_{os} →	52.9601	
	20000.0000	
	2.502842147+11	
	9.564330240+10	
	507.5132	
	80.4954	

APPENDIX II

The purpose of this appendix is to present:

(1) A full description of the geometry of each surface evaluated in this study.

(2) Friction factor (f) and Colburn Modulus (j) as a funtion of Reynolds number (RE) for each surface evaluated in this study.

(3) A complete compilation of the surface performance parameters required to compare each surface evaluated in this study.

FINNED CIRCULAR TUBES
SURFACE CF - 7.34

Tube outside diameter - 0.38 in.
Fin pitch - 7.34 per inch
Flow passage hydraulic diameter - $4r_h$=0.0154 ft.
Fin thickness (average)* - 0.018 in.
Free-flow area/frontal area - σ =0.538
Heat transfer area/total volume - α =140 ft^2/ft^3
Fin area/total area - 0.882

Note: Experimental uncertainty for heat transfer results possibly somewhat greater than the nominal ±5% quoted for the other surfaces because of the necessity of estimating a contact resistance in the bi-metal tubes.

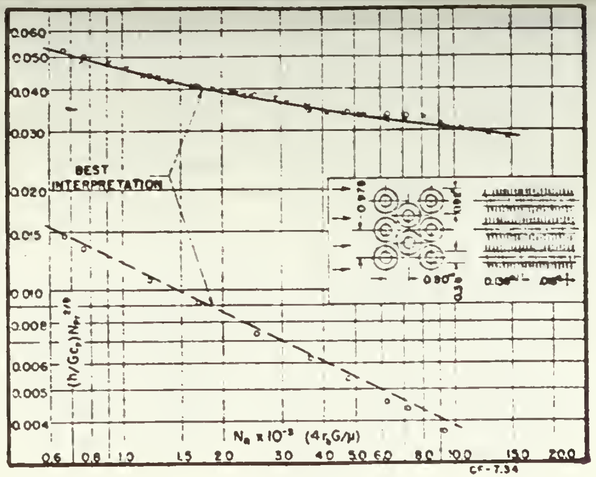
* - Fins slightly tapered.

D_N = 1.5949

$\frac{RE_N}{RE}$ = 7.6227

$\frac{f_N}{f}$ = 11.0631

$\frac{A_F b}{V_{os}}$ = 1.4282



N_R	$N_{St}N_{Pr}^{2/3}$	f
15,000	-	-
12,000	-	-
10,000	0.00388	0.0304
8,000	0.00465	0.0313
6,000	0.00500	0.0326
5,000	0.00547	0.0334
4,000	0.00609	0.0348
3,000	0.00701	0.0365
2,500	0.00770	0.0379
2,000	0.00860	0.0395
1,500	0.00990	0.0420
1,200	0.01105	0.0441
1,000	0.01210	0.0461

Heat Transfer and

Friction Data

FIG 92

APPENDIX I PROGRAM CONSTANTS						REYNOLDS NUMBER $[RE]_N$	$\frac{NTU_e}{V_{os}} = \frac{j_N RE_N A_F b}{D_N^2 V_{os}}$	$\frac{NTU_s}{V_{os}} = \frac{j_s RE A_F b}{D_N^2 V_{os}}$	$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3 A_F b}{D_N^4 V_{os}}$	$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$
0	RE	0.0000	10	A_δ/A_T	0.8420	10000	499.98	87.05	1.07 EE 11	3.48 EE 10
1	$A_{f(e)}$	0.1781	11	ℓ	0.2700	20000	710.83	132.31	7.47 EE 11	2.61 EE 11
2	$B_{f(e)}$	-0.1954	12	¶[1]	1.17 EE -04	30000	871.99	169.02	2.33 EE 12	8.48 EE 11
3	$A_{f(s)}$	0.3752	13	$A_{j(e)}$	0.3311	40000	1007.21	210.10	5.22 EE 12	1.96 EE 12
4	$B_{f(s)}$	-0.0941	14	$B_{j(e)}$	-0.4804	50000	1125.77	230.11	9.76 EE 12	3.74 EE 12
5	D_{2h}	0.1848	15	δ	0.0180	60000	1232.45	256.90	1.63 EE 13	6.35 EE 12
6	Δ_1	-0.00259	16	$A_{j(s)}$	0.5949	70000	1330.10	281.97	2.51 EE 13	9.95 EE 12
7	Δ_2	-0.00835	17	$B_{j(s)}$	-0.3960	80000	1420.60	305.65	3.65 EE 13	1.47 EE 13
8	Δ_3	-0.02426	18	Fin ID	0.0000	90000	1505.23	328.18	5.07 EE 13	2.06 EE 13
9	Δ_4	-0.45208	19	ρ	0.4130	100000	1584.95	349.76	6.82 EE 13	2.80 EE 13
A	Data ID	92.0000	D	X_L	2.1053	110000	1660.47	370.47	8.91 EE 13	3.70 EE 13
B	β	11.6666	E	X_T	2.5658	120000	1732.35	390.46	1.14 EE 14	4.76 EE 13
C	D_o	0.3800	I	Incr'mt	10000	130000	-------	------	-----------	----------
							$[1/in^2]$	$[1/in^2]$	$[1/in^4]$	$[1/in^4]$

[1]Assumes Aluminum Fin Material & Air @ 90°F

FINNED CIRCULAR TUBES
SURFACE CF - 8.72

Tube outside diameter - 0.38 in.
Fin pitch - 8.72 per inch
Flow passage hydraulic diameter - $4r_h$=0.01288 ft.
Fin thickness (average)* - 0.018 in.
Free-flow area/frontal area - σ =0.524
Heat transfer area/total volume - α =163 ft^2/ft^3
Fin area total area - 0.910

Note: Experimental uncertainty for heat transfer results possibly somewhat greater than the nominal ±5% quoted for the other surfaces because of the necessity of estimating a contact resistance in the bi-metal tubes.

* - Fins slightly tapered.

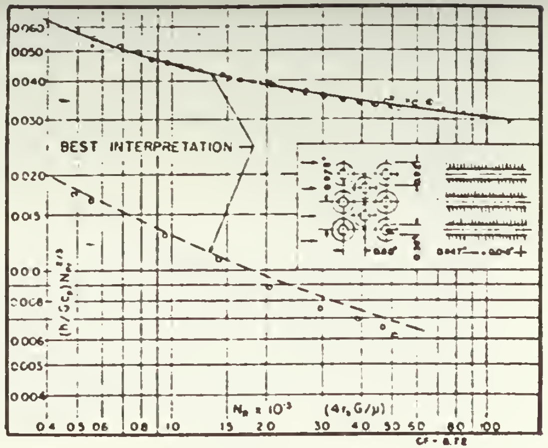


FIG. 93

N_R	$N_{St}N_{Pr}^{2/3}$	f
15,000	-	-
12,000	-	-
10,000	-	0.0306
8,000	-	0.0314
6,000	0.00645	0.0326
5,000	0.00685	0.0333
4,000	0.00740	0.0344
3,000	0.00820	0.0360
2,500	0.00880	0.0371
2,000	0.00960	0.0388
1,500	0.0108	0.0412
1,200	0.0119	0.0433
1,000	0.0129	0.0453

Heat Transfer and Friction Data

D_N = 1.5949

$\frac{f_N}{f}$ = 13.9500

$\frac{RE_N}{RE}$ = 8.8750

$\frac{A_F b}{V_{os}}$ = 1.4282

APPENDIX I PROGRAM CONSTANTS						REYNOLDS NUMBER $[RE]_N$	$\frac{NTU_e}{V_{os}} = \frac{j_N RE_N A_F b}{D_N^2 V_{os}}$	$\frac{NTU_s}{V_{os}} = \frac{j_s RE\ A_F b}{D_N^2 V_{os}}$	$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3\ A_F b}{D_N^4 V_{os}}$	$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$
0	RE	0.0000	10	A_δ/A_T	0.9100	10000	702.75	87.05	1.39 EE 11	3.48 EE 10
1	$A_{f(e)}$	0.1862	11	ℓ	0.2700	20000	1042.73	132.31	9.67 EE 11	2.61 EE 11
2	$B_{f(e)}$	-0.2016	12	τ^1	1.17 EE -04	30000	1310.10	169.02	3.00 EE 12	8.48 EE 11
3	$A_{f(s)}$	0.3752	13	$A_{j(e)}$	0.2262	40000	1538.22	201.10	6.73 EE 12	1.96 EE 12
4	$B_{f(s)}$	-0.0941	14	$B_{j(e)}$	-0.4127	50000	1740.51	230.10	1.26 EE 13	3.74 EE 12
5	D_{2h}	0.1546	15	δ	0.0180	60000	1924.06	256.90	2.09 EE 13	6.35 EE 12
6	Δ_1	-0.00259	16	$A_{j(s)}$	0.5949	70000	2093.17	281.97	3.22 EE 13	9.95 EE 12
7	Δ_2	-0.00835	17	$B_{j(s)}$	-0.3960	80000	2250.69	305.65	4.68 EE 13	1.47 EE 13
8	Δ_3	-0.02426	18	Fin ID	0.0000	90000	2398.61	328.18	6.51 EE 13	2.06 EE 13
9	Δ_4	-0.45203	19	ρ	0.4130	100000	2538.42	349.75	8.74 EE 13	2.80 EE 13
A	Data ID	93.0000	D	X_L	2.1053	110000	2671.27	370.47	1.14 EE 14	3.70 EE 13
B	β	13.5833	E	X_T	2.5658	120000	2798.01	390.46	1.46 EE 14	4.76 EE 13
C	D_o	0.380	I	Incr'mt	10000	130000	2919.37	409.81	1.82 EE 14	6.01 EE 13
							$[1/in^2]$	$[1/in^2]$	$[1/in^4]$	$[i/in^4]$

[1]Assumes Aluminum Fin Material & Air @ 90°F

FINNED CIRCULAR TUBES
SURFACE CF - 8.72(c)

Tube outside diameter - 0.42 in.
Fin pitch - 8.72 per inch
Flow passage hydraulic diameter - $4r_h$=0.01452 ft.
Fin thickness (average)* - 0.019 in.
Free-flow area/frontal area - σ =0.494
Heat transfer area/total volume - α =136 ft.^{2}/ft.^{3}
Fin area/total area - 0.876

* - Fins slightly tapered

D_N = 1.3460

$\frac{RE_N}{RE}$ = 6.6997

$\frac{f_N}{f}$ = 10.2703

$\frac{A_F b}{V_{os}}$ = 1.3844

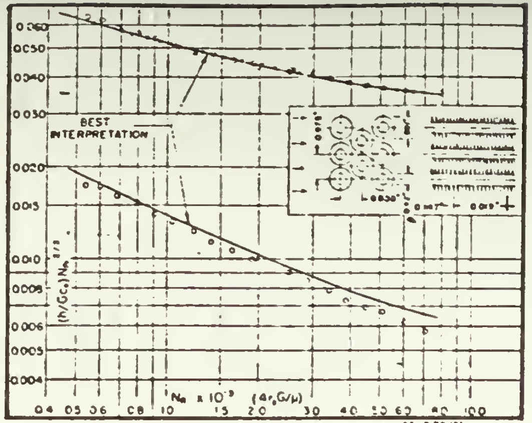


FIG. 94

N_R	$N_{St}N_{Pr}^{2/3}$	f
15,000	-	-
12,000	-	-
10,000	-	-
8,000	0.00638	0.0347
6,000	0.00685	0.0359
5,000	0.00727	0.0368
4,000	0.00765	0.0381
3,000	0.00879	0.0401
2,500	0.00945	0.0418
2,000	0.01035	0.0440
1,500	0.01170	0.0470
1,200	0.01285	0.0495
1,000	0.01390	0.0520

Heat Transfer and Friction Data

APPENDIX I PROGRAM CONSTANTS						REYNOLDS NUMBER $[RE]_N$	$\frac{NTU_e}{V_{os}} = \frac{j_N RE_N A_F b}{D_N^2 V_{os}}$	$\frac{NTU_s}{V_{os}} = \frac{j_s RE\ A_F b}{D_N^2 V_{os}}$	$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3\ A_F b}{D_N^4 V_{os}}$	$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$
0	RE	0.0000	10	A_δ/A_T	0.8760	10000	694.67	106.46	2.07 EE 11	5.65 EE 10
1	$A_{f(e)}$	0.2377	11	ℓ	0.2205	20000	1045.96	161.81	1.43 EE 12	4.25 EE 11
2	$B_{f(e)}$	-0.2192	12	ς^1	6.23 EE -05	30000	1327.51	206.71	4.40 EE 12	1.39 EE 12
3	$A_{f(s)}$	0.3053	13	$A_{j(e)}$	0.2254	40000	1571.18	245.94	9.80 EE 12	3.19 EE 12
4	$B_{f(s)}$	-0.0895	14	$B_{j(e)}$	-0.4027	50000	1789.86	281.43	1.82 EE 13	6.11 EE 12
5	D_{2h}	0.1742	15	δ	0.0190	60000	1990.36	314.19	3.03 EE 13	1.04 EE 13
6	Δ_1	-0.00259	16	$A_{j(s)}$	0.5346	70000	2176.79	344.85	4.65 EE 13	1.63 EE 13
7	Δ_2	-0.00835	17	$B_{j(s)}$	-0.3960	80000	2351.91	373.81	6.73 EE 13	2.40 EE 13
8	Δ_3	-0.02426	18	Fin ID	0.0000	90000	2517.64	401.37	9.34 EE 13	3.39 EE 13
9	Δ_4	-0.45208	19	ρ	0.4878	100000	2675.42	427.75	1.25 EE 14	4.60 EE 13
A	Data ID	94.0000	D	X_L	1.9048	110000	2826.33	453.09	1.63 EE 14	6.07 EE 13
B	β	11.3333	E	X_T	2.3214	120000	2971.22	477.54	2.08 EE 14	7.82 EE 13
C	D_o	0.420	I	Incr'mt	10000	130000	3110.80	501.20	2.60 EE 14	9.87 EE 13
							$[1/in^2]$	$[1/in^2]$	$[1/in^4]$	$[1/in^4]$

[1]Assumes Copper Fin Material & Air @ 90°F

FINNED CIRCULAR TUBES
SURFACE CF-7.0 - 5/8J
(Data of Jameson)

Tube outside diameter - 0.645 in.
Fin pitch - 7.0 per inch
Flow passage hydraulic diameter - $4r_h$=0.0219 ft.
Fin thickness - 0.010 in.
Free-flow area/frontal area - σ =0.449
Heat transfer area/total volume - α =82 ft^2/ft^3
Fin area/total area - 0.830

Note: Minimum free-flow area is in spaces transverse to flow.
These data are included in this compilation because they include compact arrangements of interest that are not adequately covered by Figs. 92 - 95.

D_N = 1.5643

$\frac{RE_N}{RE}$ = 5.6088

$\frac{f_N}{f}$ = 6.7043

$\frac{A_F b}{V_{os}}$ = 1.1859

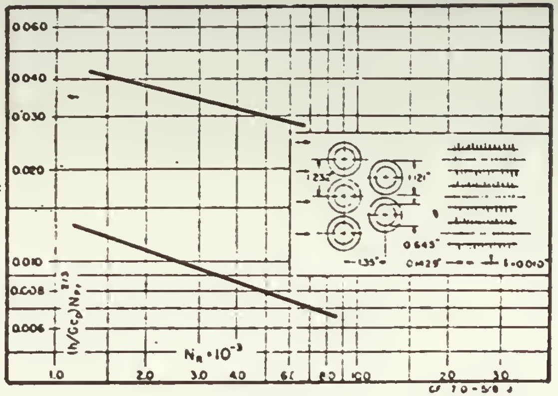


FIG. 96

NOT

AVAILABLE

Heat Transfer and

Friction Data

APPENDIX I PROGRAM CONSTANTS						REYNOLDS NUMBER $[RE]_N$	$NTU_e = \frac{j_N RE_N A_F b}{V_{cs} \; D_N^2 V_{os}}$	$NTU_s = \frac{j_s RE \; A_F b}{V_{os} \; D_N^2 V_{cs}}$	$\frac{P_e}{V_{cs}} = \frac{f_N RE_N^3 \; A_F b}{D_N^4 V_{os}}$	$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{cs}}$
0	RE	0.0000	10	A_f/A_T	0.8300	10000	319.65	57.62	5.17 EE 10	2.12 EE 10
1	$A_{f(e)}$	0.2646	11	ℓ	0.2380	20000	499.24	87.59	3.47 EE 11	1.57 EE 11
2	$B_{f(e)}$	-0.2559	12	φ[1]	6.23 EE -5	30000	646.39	111.88	1.05 EE 12	5.09 EE 11
3	$A_{f(s)}$	0.2801	13	$A_{j(e)}$	0.1461	40000	775.28	133.11	2.32 EE 12	1.17 EE 12
4	$B_{f(s)}$	-0.1046	14	$B_{j(e)}$	-0.3418	50000	891.80	152.32	4.23 EE 12	2.24 EE 12
5	D_{2h}	0.2628	15	δ	0.0100	60000	999.14	170.05	7.06 EE 12	3.79 EE 12
6	Δ_1	-0.00259	16	$A_{j(s)}$	0.4562	70000	1099.28	186.64	1.08 EE 13	5.92 EE 12
7	Δ_2	-0.00335	17	$B_{j(s)}$	-0.3960	80000	1193.53	202.32	1.55 EE 13	8.72 EE 12
8	Δ_3	-0.02426	18	Fin ID	-1.0000	90000	1282.85	217.24	2.15 EE 13	1.23 EE 13
9	Δ_4	-0.45203	19	ρ	0.5740	100000	1367.94	231.51	2.87 EE 13	1.66 EE 13
A	Data ID	96.0000	D	X_L	2.0930	110000	-------	-------	----------	----------
B	β	6.8333	E	X_T	1.9101	120000	-------	-------	----------	----------
C	D_o	0.6450	I	Incr'mt	10000	130000	-------	-------	----------	----------
							$[1/in^2]$	$[1/in^2]$	$[1/in^4]$	$[1/in^4]$

[1]Assumes Copper Fin Material & Air @ 90°F

FINNED CIRCULAR TUBES
SURFACES CF - 8.7 - 5/8J
(Data of Jameson)

Tube outside diameter - 0.645 in.
Fin pitch - 8.7 per inch
Fin thickness - 0.010 in.
Fin area/total area - 0.862

	A	B
Flow passage hydraulic diameter - $4r_h$ =	0.01797	0.0383 ft.
Free-flow area/frontal area - σ =	0.443	0.628
Heat transfer area/total volume - α =	98.7	65.7 $ft.^2/ft.^3$

Note: Minimum free-flow area is in spaces transverse to flow. These data are included in this compilation because they include compact arrangements of interest that are not adequately covered by Figs. 92 - 98.

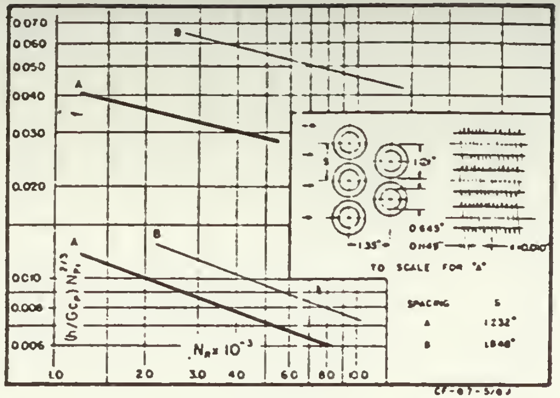


FIG. 97 A

NOT

AVAILABLE

Heat Transfer and

Friction Data

D_N = 1.5643 $\frac{f_N}{f}$ = 8.3760

$\frac{RE_N}{RE}$ = 6.7510 $\frac{A_F b}{V_{os}}$ = 1.1859

APPENDIX I PROGRAM CONSTANTS						REYNOLDS NUMBER $[RE]_N$	$\frac{NTU_e}{V_{os}} = \frac{j_N RE_N A_F b}{D_N^2 V_{os}}$	$\frac{NTU_s}{V_{os}} = \frac{j_s RE A_F b}{D_N^2 V_{os}}$	$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3 A_F b}{D_N^4 V_{os}}$	$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$
0	RE	0.0000	10	A_j/A_T	0.8620	10000	384.52	57.62	6.41 EE 10	2.12 EE 10
1	$A_{f(e)}$	0.2319	11	ℓ	0.2390	20000	593.20	87.58	4.33 EE 11	1.57 EE 11
2	$B_{f(e)}$	-0.2454	12	ϕ'[1]	6.23 EE -05	30000	762.57	111.88	1.32 EE 12	5.09 EE 11
3	$A_{f(s)}$	0.2801	13	$A_{j(e)}$	0.1541	40000	910.03	133.11	2.92 EE 12	1.17 EE 12
4	$B_{f(s)}$	-0.1046	14	$B_{j(e)}$	-0.3596	50000	1042.76	152.32	5.40 EE 12	2.24 EE 12
5	D_{2h}	0.2156	15	δ	0.0100	60000	1164.62	170.05	8.93 EE 12	3.79 EE 12
6	Δ_1	-0.00259	16	$A_{j(s)}$	0.4562	70000	1277.98	136.64	1.36 EE 13	5.92 EE 12
7	Δ_2	-0.00835	17	$B_{j(s)}$	-0.3960	80000	1384.42	202.32	1.97 EE 13	8.72 EE 12
8	Δ_3	-0.02426	18	Fin ID	-1.0000	90000	1485.09	217.24	2.73 EE 13	1.23 EE 13
9	Δ_4	-0.45208	19	ρ	0.5740	100000	1580.82	231.51	3.65 EE 13	1.66 EE 13
A	Data ID	97.1000	D	X_L	2.0930	110000	1672.27	245.23	4.74 EE 13	2.19 EE 13
B	β	8.225	E	X_T	1.9101	120000	-------	------	----------	----------
C	D_o	0.6450	I	Incr'mt	10000	130000	-------	------	----------	----------
							$[1/in^2]$	$[1/in^2]$	$[1/in^4]$	$[1/in^4]$

[1]Assumes Copper Fin Material & Air @ 90°F

FINNED CIRCULAR TUBES
SURFACES CF - 8.7 - 5/8J
(Data of Jameson)

Tube outside diameter - 0.645 in.
Fin pitch - 8.7 per inch
Fin thickness - 0.010 in.
Fin area total area - 0.862

	A	B
Flow passage hydraulic diameter - $4r_h$ =	0.01797	0.0383 ft.
Free-flow area/frontal area - σ =	0.443	0.628
Heat transfer area/total volume - α =	98.7	65.7 ft^2/ft^3

Note: Minimum free-flow area is in spaces transverse to flow. These data are included in this compilation because they include compact arrangements of interest that are not adequately covered by Figs. 92 - 98.

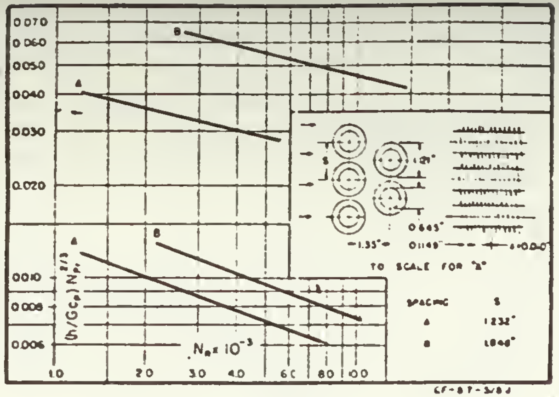


FIG. 97 B

NOT

AVAILABLE

Heat Transfer and

Friction Data

D_N = 3.2059

$\frac{f_N}{f}$ = 7.4695

$\frac{RE_N}{RE}$ = 6.7408

$\frac{A_F b}{V_{os}}$ = 1.4982

APPENDIX I PROGRAM CONSTANTS						REYNOLDS NUMBER $(RE)_N$	$\frac{NTU_e}{V_{os}} = \frac{j_N RE_N A_F b}{D_N^2 V_{os}}$	$\frac{NTU_s}{V_{os}} = \frac{j_s RE A_F b}{D_N^2 V_{os}}$	$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3 A_F b}{D_N^4 V_{os}}$	$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$
0	RE	0.0000	10	A_δ/A_T	0.8620	10000	149.17	24.64	8.13 EE 09	2.51 EE 09
1	$A_{f(e)}$	0.5416	11	ℓ	0.2380	20000	229.90	37.45	5.40 EE 10	1.87 EE 10
2	$B_{f(e)}$	-0.2676	12	¶[1]	6.23 EE -05	30000	295.64	47.84	1.64 EE 11	6.04 EE 10
3	$A_{f(s)}$	0.4639	13	$A_{j(e)}$	0.2163	40000	353.08	56.91	3.59 EE 11	1.39 EE 11
4	$B_{f(s)}$	-0.1046	14	$B_{j(e)}$	-0.3666	50000	404.97	65.13	6.60 EE 11	2.65 EE 11
5	D_{2h}	0.4596	15	δ	0.0100	60000	452.76	72.71	1.09 EE 12	4.50 EE 11
6	Δ_1	-0.00259	16	$A_{j(s)}$	0.6485	70000	497.38	79.80	1.66 EE 12	7.03 EE 11
7	Δ_2	-0.00835	17	$B_{j(s)}$	-0.3960	80000	539.40	86.51	2.39 EE 12	1.03 EE 12
8	Δ_3	-0.02426	18	Fin ID	-1.0000	90000	579.27	92.88	3.29 EE 12	1.45 EE 12
9	Δ_4	-0.45208	19	ρ	0.5754	100000	617.30	98.99	4.39 EE 12	1.97 EE 12
A	Data ID	97.2000	D	X_L	2.0930	110000	653.73	104.85	5.69 EE 12	2.60 EE 12
B	β	5.475	E	X_T	2.8651	120000	688.76	110.51	7.22 EE 12	3.35 EE 12
C	D_o	0.6450	I	Incr'mt	10000	130000	722.54	115.98	8.99 EE 12	4.22 EE 12
							$[1/in^2]$	$[1/in^2]$	$[1/in^4]$	$[1/in^4]$

[1]Assumes Copper Fin Material & Air @ 90°F

FINNED CIRCULAR TUBES
SURFACES CF - 9.05-3/4J
(Data of Jameson)
Tube outside diameter - 0.774 in.
Fin pitch - 9.05 per inch
Fin thickness - 0.012 in.
Fin area/total area - 0.835

	A	B
Flow passage hydraulic diameter - $4r_h$ =	0.01681	0.02685
Free-flow area/frontal area - σ =	0.455	0.572
Heat transfer area/total volume - α =	108	85.1

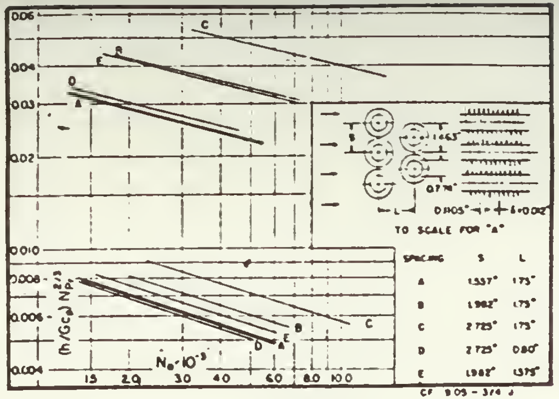


FIG. 98 A

NOT

AVAILABLE

Heat Transfer and

Friction Data

D_N = 2.2541

$\frac{f_N}{f}$ = 13.7185

$\frac{RE_N}{RE}$ = 10.0851

$\frac{A_F b}{V_{os}}$ = 1.2157

APPENDIX I PROGRAM CONSTANTS						REYNOLDS NUMBER $[RE]_N$	$\frac{NTU_e}{V_{cs}} = \frac{j_N RE_N A_F b}{D_N^2 V_{os}}$	$\frac{NTU_s}{V_{os}} = \frac{j_s RE\ A_F b}{D_N^2 V_{os}}$	$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3\ A_F b}{D_N^4 V_{os}}$	$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$
0	RE	0.0000	10	A_f/A_T	0.8350	10000	230.42	30.46	2.26 EE 10	5.64 EE 09
1	$A_{f(e)}$	0.2263	11	ℓ	0.3445	20000	363.32	46.29	1.50 EE 11	4.24 EE 10
2	$B_{f(e)}$	-0.2707	12	¶[1]	6.23 EE -05	30000	472.95	59.14	4.53 EE 11	1.38 EE 11
3	$A_{f(s)}$	0.2753	13	$A_{j(e)}$	0.0342	40000	569.34	70.36	9.93 EE 11	3.18 EE 11
4	$B_{f(s)}$	-0.0904	14	$B_{j(e)}$	-0.3273	50000	656.71	80.51	1.83 EE 12	6.10 EE 11
5	D_{2h}	0.2017	15	δ	0.0120	60000	737.36	89.89	3.00 EE 12	1.04 EE 12
6	Δ_1	-0.00259	16	$A_{j(s)}$	0.4884	70000	812.71	98.66	4.57 EE 12	1.62 EE 12
7	Δ_2	-0.00835	17	$B_{j(s)}$	-0.3960	80000	883.70	106.94	6.58 EE 12	2.39 EE 12
8	Δ_3	-0.02426	18	Fin ID	-1.0000	90000	951.05	114.83	9.08 EE 12	3.37 EE 12
9	Δ_4	-0.45208	19	ρ	0.5290	100000	1015.25	122.37	1.21 EE 13	4.58 EE 12
A	Data ID	98.10000	D	X_L	2.2610	110000	1076.71	129.62	1.57 EE 13	6.04 EE 12
B	β	9.0000	E	X_T	2.0116	120000	1135.74	136.62	1.99 EE 13	7.79 EE 12
C	D_o	0.7740	I	Incr'mt	10000	130000	1192.60	143.39	2.48 EE 13	9.83 EE 12
							$[1/in^2]$	$[1/in^2]$	$[1/in^4]$	$[1/in^4]$

[1]Assumes Copper Fin Material & Air @ 90°F

FINNED CIRCULAR TUBES
SURFACES CF - 9.05-3/4J
(Data of Jameson)
Tube outside diameter - 0.774 in.
Fin pitch - 9.05 per inch
Fin thickness - 0.012 in.
Fin area/total area - 0.835

	A	B
Flow passage hydraulic diameter - $4r_h$ =	0.01681	0.02685
Free-flow area/frontal area - σ =	0.455	0.572
Heat transfer area/total volume - α =	108	85.1

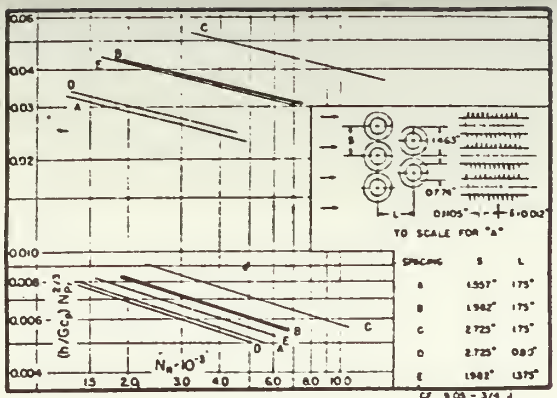


FIG. 98 B

NOT

AVAILABLE

Heat Transfer and

Friction Data

D_N = 3.4776

$\frac{f_N}{f}$ = 12.2870

$\frac{RE_N}{RE}$ = 10.1158

$\frac{A_F b}{V_{os}}$ = 1.4103

APPENDIX I PROGRAM CONSTANTS

0	RE	0.0000	10	A_δ/A_T	0.8350
1	$A_{f(e)}$	0.2905	11	ℓ	0.3445
2	$B_{f(e)}$	-0.2531	12	$\P^1$	6.23 EE -05
3	$A_{f(s)}$	0.3775	13	$A_{j(e)}$	0.0947
4	$B_{f(s)}$	-0.0904	14	$B_{j(e)}$	-0.3239
5	D_{2h}	0.3222	15	δ	0.0120
6	Δ_1	-0.00259	16	$A_{j(s)}$	0.6037
7	Δ_2	-0.00835	17	$B_{j(s)}$	-0.3960
8	Δ_3	-0.02426	18	Fin ID	-1.0000
9	Δ_4	-0.45208	19	o	0.5290
A	Data ID	98.2000	D	X_L	2.2610
B	β	7.09166	E	X_T	2.5607
C	D_o	0.774	I	Incr'mt	10000

REYNOLDS NUMBER $[RE]_N$	$\frac{NTU_e}{V_{os}} = \frac{j_N RE_N A_F b}{D_N^2 V_{os}}$	$\frac{NTU_s}{V_{os}} = \frac{j_s RE\ A_F b}{D_N^2 V_{os}}$	$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3\ A_F b}{D_N^4 V_{os}}$	$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$
10000	125.61	18.35	6.01 EE 09	1.57 EE 09
20000	199.11	27.89	4.03 EE 10	1.18 EE 10
30000	260.17	35.63	1.23 EE 11	3.83 EE 10
40000	314.16	42.39	2.71 EE 11	8.84 EE 10
50000	363.35	48.50	5.00 EE 11	1.69 EE 11
60000	408.95	54.15	8.25 EE 11	2.88 EE 11
70000	451.73	59.43	1.26 EE 12	4.51 EE 11
80000	492.18	64.43	1.82 EE 12	6.65 EE 11
90000	530.68	69.18	2.51 EE 12	9.36 EE 11
100000	567.51	73.72	3.35 EE 12	1.27 EE 12
110000	602.88	78.09	4.36 EE 12	1.68 EE 12
120000	636.95	82.30	5.53 EE 12	2.16 EE 12
130000	669.85	86.38	6.90 EE 12	2.73 EE 12
	$[1/in^2]$	$[1/in^2]$	$[1/in^4]$	$[1/in^4]$

[1]Assumes Copper Fin Material & Air @ 90°F

FINNED CIRCULAR TUBES
SURFACES CF - 9.05-3/4J
(Data of Jameson)
Tube outside diameter - 0.774 in.
Fin pitch - 9.05 per inch
Fin thickness - 0.012 in.
Fin area/total area - 0.835

	C	D	E
Flow passage hydraulic diameter - $4r_h$ =	0.0445	0.01587	0.02108 ft.
Free-flow area/frontal area - σ =	0.688	0.537	0.572
Heat transfer area/total volume - α =	61.9	135	108 ft^2/ft^3

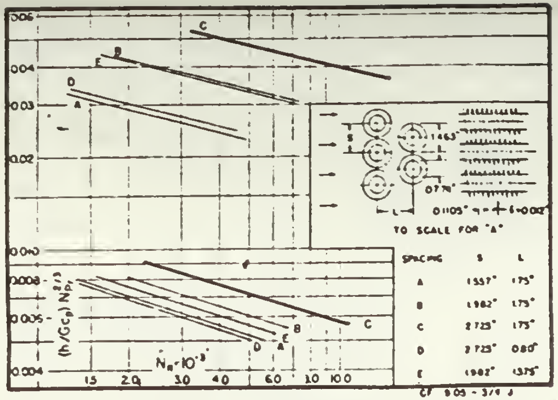


FIG. 9.1 C

NOT

AVAILABLE

Heat Transfer and

Friction Data

D_N = 5.6165

$\frac{f_N}{f}$ = 11.3690

$\frac{RE_N}{RE}$ = 10.1163

$\frac{A_F b}{V_{os}}$ = 1.5887

APPENDIX I PROGRAM CONSTANTS

0	RE	0.0000	10	A_f/A_T	0.8350
1	$A_{f(e)}$	0.3850	11	ℓ	0.3145
2	$B_{f(e)}$	-0.2462	12	q[1]	6.23 EE -05
3	$A_{f(s)}$	0.5232	13	$A_{j(e)}$	0.1023
4	$B_{f(s)}$	-0.0904	14	$B_{j(e)}$	-0.3137
5	D_{2h}	0.5340	15	δ	0.0120
6	Δ_1	-0.00259	16	$A_{j(s)}$	0.7697
7	Δ_2	-0.00835	17	$B_{j(s)}$	-0.3960
8	Δ_3	-0.02426	18	Fin ID	-1.0000
9	Δ_4	-0.45208	19	ρ	0.5290
A	Data ID	98.3000	D	X_L	2.2610
B	β	5.1583	E	X_T	3.5207
C	D_o	0.774	I	Incr'mt	10000

REYNOLDS NUMBER $[RE]_n$	$\frac{NTU_e}{V_{cs}} = \frac{j_N RE_N A_F b}{D_N^2 V_{os}}$	$\frac{NTU_s}{V_{os}} = \frac{j_s RE A_F b}{D_N^2 V_{os}}$	$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3 A_F b}{D_N^4 V_{os}}$	$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$
10000	61.56	10.10	1.28 EE 09	3.63 EE 08
20000	98.49	15.36	8.63 EE 09	2.73 EE 09
30000	129.47	19.62	2.63 EE 10	8.83 EE 09
40000	157.06	23.34	5.82 EE 10	2.05 EE 10
50000	182.33	26.71	1.08 EE 11	3.93 EE 10
60000	205.38	29.81	1.78 EE 11	6.67 EE 10
70000	228.05	32.72	2.72 EE 11	1.05 EE 11
80000	249.11	35.47	3.92 EE 11	1.54 EE 11
90000	269.23	38.09	5.43 EE 11	2.17 EE 11
100000	288.54	40.59	7.25 EE 11	2.95 EE 11
110000	307.13	43.00	9.43 EE 11	3.89 EE 11
120000	325.10	45.32	1.20 EE 12	5.01 EE 11
130000	342.50	47.56	1.49 EE 12	6.33 EE 11
	$[1/in^2]$	$[1/in^2]$	$[1/in^4]$	$[1/in^4]$

[1]Assumes Copper Fin Material & Air @ 90°F

FINNED CIRCULAR TUBES
SURFACES CF - 9.05-3/4J
(Data of Jameson)
Tube outside diameter - 0.774 in.
Fin pitch - 9.05 per inch
Fin thickness - 0.012 in.
Fin area/total area - 0.830

	C	B	E
Flow passage hydraulic diameter - $4r_h$ =	0.0445	0.01567	0.02104 ft.
Free-flow area/frontal area - σ =	0.606	0.627	0.572
Heat transfer area/total volume - α =	61.0	138	108 ft^2/ft^3

D_N = 2.1214

$\frac{f_N}{f}$ = 13.5882

$\frac{RE_N}{RE}$ = 10.0860

$\frac{A_F b}{V_{os}}$ = 1.5332

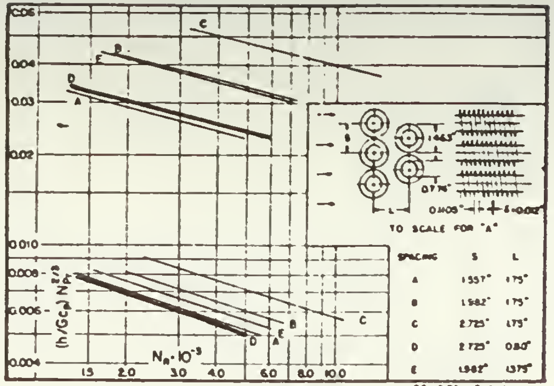


FIG. 99 D

NOTE: Minimum free flow area is in diagonals

NOT

AVAILABLE

Heat Transfer and

Friction Data

APPENDIX I PROGRAM CONSTANTS						REYNOLDS NUMBER $[RE]_N$	$\frac{NTU_e}{V_{os}} = \frac{j_N RE_N A_F b}{D_N^2 V_{os}}$	$\frac{NTU_s}{V_{os}} = \frac{j_s RE\ A_F b}{D_N^2 V_{os}}$	$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3 A_F b}{D_N^4 V_{os}}$	$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$
0	X_D	2.0413	10	A_δ/A_T	0.8350	10000	320.22	52.96	3.74 EE 10	1.37 EE 10
1	$A_{f(e)}$	0.2173	11	$\ell/\sqrt{\delta}$	3.1448	20000	507.51	80.49	2.50 EE 11	9.56 EE 10
2	$B_{f(e)}$	-0.2590	12	¶[1]	6.23 EE -05	30000	663.40	102.83	7.61 EE 11	2.98 EE 11
3	$A_{f(s)}$	1.1125	13	$A_{j(e)}$	0.0813	40000	801.54	122.35	1.67 EE 12	6.67 EE 11
4	$B_{f(s)}$	-0.1971	14	$B_{j(e)}$	-0.3264	50000	927.64	140.40	3.08 EE 12	1.25 EE 12
5	D_{2h}	0.1904	15	X_T	3.5207	60000	1044.81	156.30	5.08 EE 12	2.08 EE 12
6	Δ_1	-0.00239	16	$A_{j(s)}$	0.5965	70000	1154.96	171.55	7.76 EE 12	3.20 EE 12
7	Δ_2	-0.00835	17	$B_{j(s)}$	-0.3960	80000	1259.38	185.96	1.12 EE 13	4.66 EE 12
8	Δ_3	-0.02426	18	Fin ID	-1.0000	90000	1358.98	199.67	1.54 EE 13	6.48 EE 12
9	Δ_4	-0.45208	19	ρ	0.5290	100000	1454.47	212.79	2.06 EE 13	8.71 EE 12
A	Data ID	98.4000	D	X_L	1.03359	110000	1546.37	225.40	2.68 EE 13	1.14 EE 13
B	β	11.2500	Σ	α	59.5804	120000	1635.10	236.56	3.40 EE 13	1.45 EE 13
C	D_o	0.7740	I	Incr'mt	10000	130000	1721.01	249.33	4.23 EE 13	1.82 EE 13
							$[1/in^2]$	$[1/in^2]$	$[1/in^4]$	$[1/in^4]$

[1]Assumes Copper Fin Material & Air @ 90°F

FINNED CIRCULAR TUBES
SURFACES CF - 9.05-3/4J
(Data of Jameson)

Tube outside diameter - 0.774 in.
Fin pitch - 9.05 per inch
Fin thickness - 0.012 in.
Fin area/total area - 0.838

	C	D	E
Flow passage hydraulic diameter - $4r_h$ =	0.0445	0.01587	0.02108 ft.
Free-flow area/frontal area - σ =	0.688	0.637	0.572
Heat transfer area/total volume - α =	61.9	135	108 $ft.^2/ft.^3$

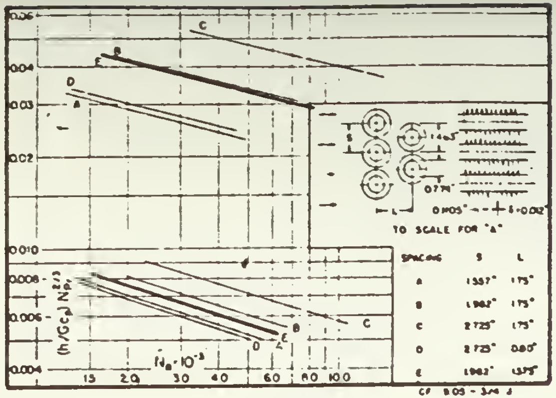


FIG. 98E

NOT

AVAILABLE

Heat Transfer and

Friction Data

D_N = 2.7324

$\frac{f_N}{f}$ = 12.3864

$\frac{RE_N}{RE}$ = 10.0869

$\frac{A_F b}{V_{os}}$ = 1.4733

APPENDIX I PROGRAM CONSTANTS						REYNOLDS NUMBER [RE]n	$\frac{NTU_e}{V_{os}} = \frac{j_N RE_N A_F b}{D_N^2 V_{os}}$	$\frac{NTU_s}{V_{os}} = \frac{j_s RE A_F b}{D_N^2 V_{os}}$	$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3 A_F b}{D_N^4 V_{os}}$	$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$
0	RE	0.0000	10	A_δ/A_T	0.8350	10000	198.12	29.38	1.64 EE 10	3.80 EE 09
1	$A_{f(e)}$	0.3020	11	ℓ	0.3445	20000	312.56	44.65	1.09 EE 11	2.77 EE 10
2	$B_{f(e)}$	-0.2607	12	q^1	6.23 EE -05	30000	407.15	57.05	3.32 EE 11	9.84 EE 10
3	$A_{f(s)}$	0.4972	13	$A_{j(e)}$	0.0915	40000	490.48	67.87	7.30 EE 11	2.01 EE 11
4	$B_{f(s)}$	-0.1349	14	$B_{j(e)}$	-0.3288	50000	566.15	77.66	1.34 EE 12	3.82 EE 11
5	D_{2h}	0.2530	15	δ	0.0120	60000	636.11	86.71	2.22 EE 12	6.44 EE 11
6	Δ_1	-0.00259	16	$A_{j(s)}$	0.5712	70000	701.57	95.17	3.38 EE 12	1.00 EE 12
7	Δ_2	-0.00835	17	$B_{j(s)}$	-0.3960	80000	763.35	103.16	4.87 EE 12	1.47 EE 12
8	Δ_3	-0.02426	18	Fin ID	-1.0000	90000	822.04	110.77	6.73 EE 12	2.06 EE 12
9	Δ_4	-0.45208	19	ρ	0.5290	100000	878.06	118.04	8.98 EE 12	2.78 EE 12
A	Data ID	98.5000	D	X_L	1.7765	110000	931.76	125.04	1.17 EE 13	3.66 EE 12
B	β	9.0000	E	X_T	2.5607	120000	983.41	131.79	1.48 EE 13	4.79 EE 12
C	D_o	0.774	I	Incr'mt	10000	130000	1033.22	138.31	1.84 EE 13	5.90 EE 12
							[$1/in^2$]	[$1/in^2$]	[$1/in^4$]	[$1/in^4$]

[1]Assumes Copper Fin Material & Air @ 90°F

FINNED CIRCULAR TUBES
SURFACES CF - 8.8 - 1.0J
(Data of Jameson)

Tube outside diameter - 1.024 in.
Fin pitch - 8.8 per inch
Fin thickness - 0.012 in.
Fin area/total area - 0.825

	A	B
Flow passage hydraulic diameter - $4r_h$ =	0.01927	0.0443 ft.
Free-flow/frontal area - σ =	0.439	0.643
Heat transfer area/total volume - α =	91.2	58.1 ft^2/ft^3

Note: Minimum free-flow area is in spaces transverse to flow. These data are included in this compilation because they include compact arrangements of interest that are not adequately covered by Figs. 92 - 95.

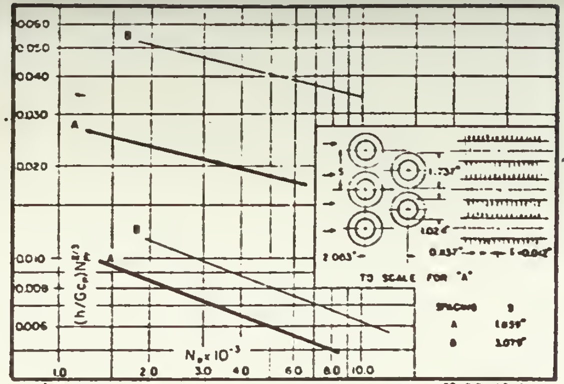


FIG. 99 A

NOT

AVAILABLE

Heat Transfer and

Friction Data

D_N = 2.3984

$\frac{f_N}{f}$ = 12.2399

$\frac{RE_N}{RE}$ = 9.5477

$\frac{A_F b}{V_{os}}$ = 1.1989

APPENDIX I PROGRAM CONSTANTS						REYNOLDS NUMBER [RE]n	$\frac{NTU_e}{V_{os}} = \frac{j_N RE_N A_F b}{D_N^2 V_{os}}$	$\frac{NTU_s}{V_{os}} = \frac{j_s RE\ A_F b}{D_N^2 V_{os}}$	$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3\ A_F b}{D_N^4 V_{os}}$	$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$
0	RE	0.0000	10	A_δ/A_T	0.8250	10000	226.75	24.60	1.22 EE 10	3.82 EE 09
1	$A_{f(e)}$	0.1542	11	ℓ	0.3565	20000	346.34	37.39	8.23 EE 10	2.85 EE 10
2	$B_{f(e)}$	-0.2473	12	ψ[1]	6.23 EE -05	30000	442.58	47.77	2.51 EE 11	9.22 EE 10
3	$A_{f(s)}$	0.2719	13	$A_{j(e)}$	0.1432	40000	525.90	56.84	5.54 EE 11	2.12 EE 11
4	$B_{f(s)}$	-0.1028	14	$B_{j(e)}$	-0.3727	50000	600.59	65.04	1.02 EE 12	4.05 EE 11
5	D_{2h}	0.2312	15	δ	0.0120	60000	668.92	72.61	1.69 EE 12	6.87 EE 11
6	Δ_1	-0.00259	16	$A_{j(s)}$	0.4529	70000	732.30	79.69	2.59 EE 12	1.07 EE 12
7	Δ_2	-0.00835	17	$B_{j(s)}$	-0.3960	80000	791.67	86.39	3.73 EE 12	1.58 EE 12
8	Δ_3	-0.02426	18	Fin ID	-1.0000	90000	847.70	92.76	5.16 EE 12	2.22 EE 12
9	Δ_4	-0.45208	19	ρ	0.5895	100000	900.88	98.85	6.90 EE 12	3.02 EE 12
A	Data ID	99.100	D	X_L	2.0146	110000	951.59	104.71	8.97 EE 12	3.98 EE 12
B	β	7.6000	E	X_T	1.9131	120000	1000.13	110.36	1.14 EE 13	5.12 EE 12
C	D_o	1.024	I	Incr'mt	10000	130000	1046.74	115.82	1.42 EE 13	6.45 EE 12
							$[1/in^2]$	$[1/in^2]$	$[1/in^4]$	$[1/in^4]$

[1]Assumes Copper Fin Material & Air @ 90°F

FINNED CIRCULAR TUBES
SURFACES CF - 8.8 - 1.0J
(Data of Jameson)
Tube outside diameter - 1.024 in.
Fin pitch - 8.8 per inch
Fin thickness - 0.012 in.
Fin area/total area - 0.825

	A	B
Flow passage hydraulic diameter - $4r_h$ =	0.01927	0.0443 ft.
Free-flow/frontal area - σ =	0.439	0.643
Heat transfer area/total volume - α =	91.2	58.1 ft^2/ft^3

Note: Minimum free-flow area is in spaces transverse to flow. These data are included in this compilation because they include compact arrangements of interest that are not adequately covered by Figs. 92 - 95.

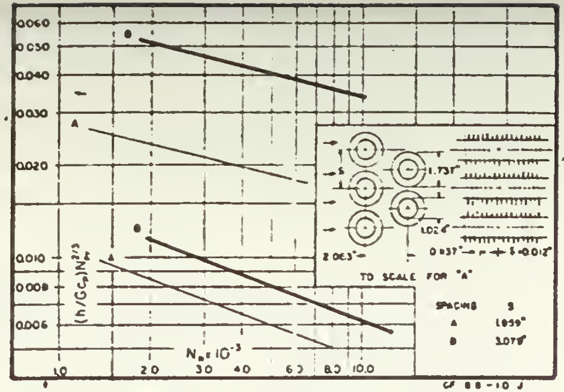


FIG 99 B

NOT

AVAILABLE

Heat Transfer and

Friction Data

D_N = 5.2713

$\frac{f_N}{f}$ = 10.6684

$\frac{RE_N}{RE}$ = 9.5599

$\frac{A_F b}{V_{os}}$ = 1.534

APPENDIX I PROGRAM CONSTANTS						REYNOLDS NUMBER $[RE]_n$	$\frac{NTU_e}{V_{os}} = \frac{j_N RE_N A_F b}{D_N^2 V_{os}}$	$\frac{NTU_s}{V_{os}} = \frac{j_s RE\ A_F b}{D_N^2 V_{os}}$	$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3 A_F b}{D_N^4 V_{os}}$	$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$
0	RE	0.0000	10	A_f/A_T	0.8250	10000	78.61	9.60	1.23 EE 09	3.65 EE 08
1	$A_{f(e)}$	0.3599	11	ℓ	0.3565	20000	120.12	14.59	8.60 EE 09	2.72 EE 09
2	$B_{f(e)}$	-0.2563	12	$\P$[1]	6.23 EE -05	30000	153.70	18.63	2.62 EE 10	8.80 EE 09
3	$A_{f(s)}$	0.4730	13	$A_{j(e)}$	0.2027	40000	182.91	22.17	5.76 EE 10	2.02 EE 10
4	$B_{f(s)}$	-0.1027	14	$B_{j(e)}$	-0.3787	50000	209.22	25.37	1.06 EE 11	3.86 EE 10
5	D_{2h}	0.5316	15	δ	0.0120	60000	233.39	28.32	1.75 EE 11	6.55 EE 10
6	Δ_1	-0.00259	16	$A_{j(s)}$	0.6671	70000	255.90	31.08	2.67 EE 11	1.02 EE 11
7	Δ_2	-0.00835	17	$B_{j(s)}$	-0.3960	80000	277.07	33.70	3.86 EE 11	1.51 EE 11
8	Δ_3	-0.02426	18	Fin ID	-1.0000	90000	297.13	36.18	5.33 EE 11	2.12 EE 11
9	Δ_4	-0.45208	19	ρ	0.5995	100000	316.23	38.56	7.11 EE 11	2.88 EE 11
A	Data ID	99.200	D	X_L	2.0146	110000	334.50	40.84	9.24 EE 11	3.79 EE 11
B	β	4.9416	E	X_T	3.0068	120000	352.06	43.05	1.17 EE 12	4.38 EE 11
C	D_o	1.024	I	Incr'mt	10000	130000	368.97	45.18	1.46 EE 12	6.16 EE 11
							$[1/in^2]$	$[1/in^2]$	$[1/in^4]$	$[1/in^4]$

[1]Assumes Copper Fin Material & Air @ 90°F

FINNED CIRCULAR TUBES
SURFACE 8.0 - 3/8 T
(Data of Trane Co.)

Tube outside diameter - 0.402 in.
Fin pitch - 8.0 per inch
Flow passage hydraulic diameter - $4r_h$=0.01192 ft.
Fin thickness - 0.013 in.
Free-flow area/frontal area - σ =0.534
Heat transfer area/total volume - α =179 ft^2/ft^3
Fin area/total area - 0.839

Note: Minimum free-flow area is spaces transverse to flow.
These data are included in this compilation because they apply to a compact surface configuration of considerable technical interest for which no data have been obtained on this project.

D_N = 1.6402

$\frac{f_N}{f}$ = 14.4115

$\frac{RE_N}{RE}$ = 10.2285

$\frac{A_F b}{V_{os}}$ = 1.4014

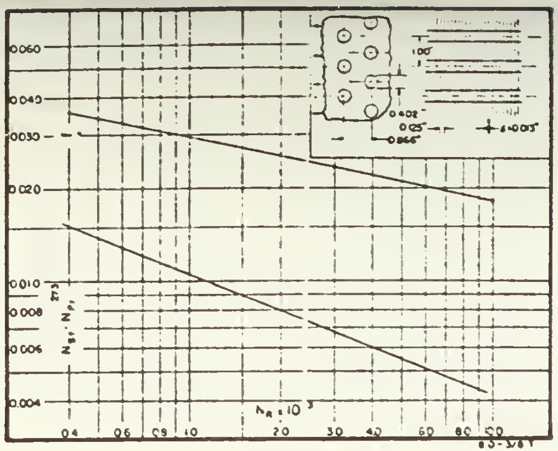


FIG 100

NOT AVAILABLE

Heat Transfer and Friction Data

APPENDIX I PROGRAM CONSTANTS

0	RE	0.00	10	A_f/A_T	0.8390
1	$A_{f(e)}$	0.1223	11	ℓ	0.3240
2	$B_{f(e)}$	-0.2073	12	η^1	1.2 EE -04
3	$A_{f(s)}$	0.3514	13	$A_{j(e)}$	0.1650
4	$B_{f(s)}$	-0.0896	14	$B_{j(e)}$	-0.3986
5	D_{2h}	0.1430	15	δ	0.0130
6	Δ_1	-0.00259	16	$A_{j(s)}$	0.5830
7	Δ_2	-0.00835	17	$B_{j(s)}$	-0.3960
8	Δ_3	-0.02426	18	Fin ID	-1.0000
9	Δ_4	-0.45208	19	ρ	0.3829
A	Data ID	100.00	D	X_L	2.1542
B	β	14.9166	E	X_T	2.4876
C	D_o	0.402	I	Incr'mt	10000

REYNOLDS NUMBER $[RE]_N$	$\frac{NTU_e}{V_{cs}} = \frac{j_N RE_N A_F b}{D_N^2 V_{os}}$	$\frac{NTU_s}{V_{os}} = \frac{j_s RE\ A_F b}{D_N^2 V_{os}}$	$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3\ A_F b}{D_N^4 V_{os}}$	$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$
10000	598.03	79.14	8.19 EE 10	2.98 EE 10
20000	883.32	120.29	5.67 EE 11	2.24 EE 11
30000	1103.45	153.67	1.76 EE 12	7.30 EE 11
40000	1282.99	182.83	3.93 EE 12	1.69 EE 12
50000	1449.00	209.21	7.33 EE 12	3.23 EE 12
60000	1592.89	233.57	1.22 EE 13	5.49 EE 12
70000	1723.54	256.36	1.88 EE 13	8.59 EE 12
80000	1843.57	277.89	2.74 EE 13	1.27 EE 13
90000	1954.80	298.38	3.78 EE 13	1.79 EE 13
100000	2058.60	317.99	5.08 EE 13	2.43 EE 13
110000	2155.99	336.83	6.63 EE 13	3.20 EE 13
120000	2247.82	355.00	8.45 EE 13	4.12 EE 13
130000	2334.72	372.59	1.06 EE 14	5.21 EE 13
	$[1/in^2]$	$[1/in^2]$	$[1/in^4]$	$[1/in^4]$

[1]Assumes Aluminum Fin Material & Air @ 90°F

FINNED CIRCULAR TUBES
SURFACE 7.75 - 5/8 T
(Data of Trane Co.)

Tube outside diameter - 0.676 in.
Fin pitch - 7.75 per inch
Flow passage hydraulic diameter - $4r_h$ = 0.01268 ft.
Fin thickness - 0.016 in.
Free-flow area/frontal area - σ = 0.497
Heat transfer area/total volume - α = 157 $ft.^2/ft.^3$
Fin area total area - 0.905

Note: Minimum free-flow area in spaces transverse to flow.
These data are included in this compilation because they apply to a compact surface configuration of considerable technical interest for which no data have been obtained on this project.

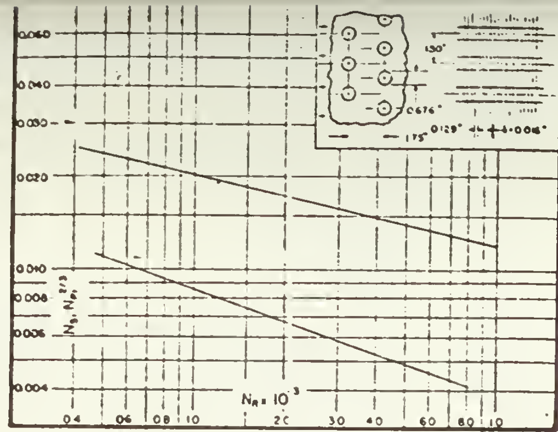


FIG 101

NOT

AVAILABLE

Heat Transfer and

Friction Data

D_N = 2.7160 $\frac{f_N}{f}$ = 21.7461

$\frac{RE_N}{RE}$ = 16.1715 $\frac{A_F b}{V_{os}}$ = 1.2727

APPENDIX I PROGRAM CONSTANTS						REYNOLDS NUMBER $[RE]_N$	$\frac{NTU_e}{V_{cs}} = \frac{j_N RE_N A_F b}{D_N^2 V_{cs}}$	$\frac{NTU_s}{V_{os}} = \frac{j_s RE\, A_F b}{D_N^2 V_{os}}$	$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3 A_F b}{D_N^4 V_{cs}}$	$\frac{F_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$
0	RE	0.00	10	A_δ/A_T	0.905	10000	277.87	24.79	1.14 EE 10	3.44 EE 09
1	$A_{f(e)}$	0.0966	11	ℓ	0.6616	20000	410.58	37.68	7.81 EE 10	2.66 EE 10
2	$B_{f(e)}$	-0.2270	12	q^1	1.2 EE -4	30000	508.56	48.14	2.40 EE 11	8.77 EE 10
3	$A_{f(s)}$	0.2408	13	$A_{j(e)}$	0.0940	40000	587.05	57.27	5.34 EE 11	2.05 EE 11
4	$B_{f(s)}$	-0.0534	14	$B_{j(e)}$	-0.3458	50000	652.48	65.54	9.92 EE 11	3.95 EE 11
5	D_{2h}	0.1522	15	δ	0.0160	60000	708.37	73.16	1.64 EE 12	6.76 EE 11
6	Δ_1	-0.00259	16	$A_{j(s)}$	0.5514	70000	756.38	80.30	2.52 EE 12	1.06 EE 12
7	Δ_2	-0.00835	17	$B_{j(s)}$	-.3960	80000	799.47	87.05	3.65 EE 12	1.58 EE 12
8	Δ_3	-0.02426	18	Fin ID	-1.000	90000	837.17	93.47	5.06 EE 12	2.23 EE 12
9	Δ_4	-0.45203	19	ρ	0.3381	100000	870.76	99.61	6.73 EE 12	3.05 EE 12
A	Data ID	101.00	D	X_L	2.5888	110000	900.84	105.51	8.33 EE 12	4.03 EE 12
B	β	13.083	E	X_T	2.2189	120000	927.87	111.20	1.12 EE 13	5.21 EE 12
C	D_o	0.676	I	Incr'mt	10000	130000	952.22	116.71	1.40 EE 13	6.60 EE 12
							$[1/in^2]$	$[1/in^2]$	$[1/in^4]$	$[1/in^4]$

[1]Assumes Aluminum Fin Material & Air @ 90°F

FINNED CIRCULAR TUBES[1]
(data of Trane Co.)

Tube outside Dia: 0.391 in
Fin Pitch: 8.0 per in
Flow Pass. Hyd. Dia.: .01268 ft.
Fin Thickness: 0.008 in
Total Transfer area/Tot. Vol: 180 ft^2/ft^3
Fin area/Tot. Area: .916

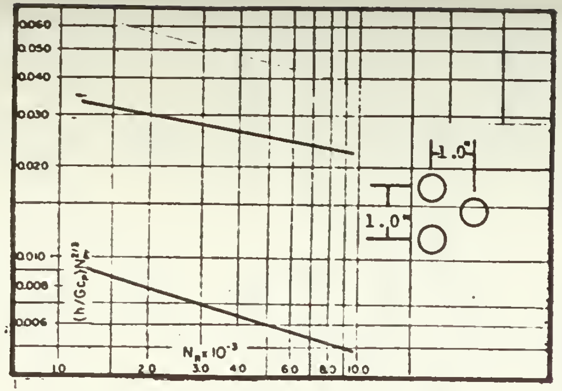


FIG 102 SPECIAL

NOT

AVAILABLE

Heat Transfer and

Friction Data

D_N = 1.7174 $\frac{f_N}{f}$ = 12.8670

$\frac{RE_N}{RE}$ = 10.5751 $\frac{A_F b}{V_{os}}$ = 1.4141

APPENDIX I PROGRAM CONSTANTS						REYNOLDS NUMBER $[RE]_N$	$\frac{NTU_e}{V_{os}} = \frac{j_N RE_N A_F b}{D_N^2 V_{os}}$	$\frac{NTU_s}{V_{os}} = \frac{j_s RE\ A_F b}{D_N^2 V_{os}}$	$\frac{P_e}{V_{os}} = \frac{f_N RE_N^3\ A_F b}{D_N^4 V_{os}}$	$\frac{P_s}{V_{os}} = \frac{f_s RE^3 A_F b}{D_N^4 V_{os}}$
0	RE	0.0000	10	A_f/A_T	0.9160	10000	510.07	75.00	7.57 EE 10	2.62 EE 10
1	$A_{f(e)}$	0.1367	11	ℓ	0.3295	20000	802.42	114.00	5.30 EE 11	1.98 EE 11
2	$B_{f(e)}$	-0.1938	12	ϵ^1	6.23 EE -05	30000	1038.78	145.63	1.65 EE 12	6.46 EE 11
3	$A_{f(s)}$	0.3501	13	$A_{j(e)}$	0.0814	40000	1242.49	173.27	3.70 EE 12	1.49 EE 12
4	$B_{f(s)}$	-0.0841	14	$B_{j(e)}$	-0.3073	50000	1423.59	198.27	6.93 EE 12	2.8' EE 12
5	D_{2h}	0.1522	15	δ	0.0080	60000	1587.63	221.35	1.16 EE 13	4.87 EE 12
6	Δ_1	-0.00259	16	$A_{j(s)}$	0.6003	70000	1738.10	242.95	1.78 EE 13	7.64 EE 12
7	Δ_2	-0.00835	17	$B_{j(s)}$	-0.3960	80000	1877.38	263.35	2.59 EE 13	1.13 EE 13
8	Δ_3	-0.02426	18	Fin ID	-1.0000	90000	2007.21	282.77	3.61 EE 13	1.59 EE 13
9	Δ_4	-0.45208	19	ρ	0.3724	100000	2128.89	301.35	4.85 EE 13	2.16 EE 13
A	Data ID	102 SPECIAL	D	X_L	2.2148	110000	2243.43	319.21	6.33 EE 13	2.85 EE 13
B	β	15.0000	E	X_T	2.5575	120000	2351.66	336.43	8.08 EE 13	3.68 EE 13
C	D_o	0.3910	I	Incr'mt	10000	130000	2454.24	353.10	1.01 EE 14	4.64 EE 14
							$[1/in^2]$	$[1/in^2]$	$[1/in^4]$	$[1/in^4]$

[1] Assumes Copper Fin Material & Air @ 90°F

APPENDIX III

SIZING CROSSFLOW FINNED TUBULAR HEAT EXCHANGERS FOR A GIVEN JOB

Crossflow finned tubular heat exchangers have found application as gas turbine plant intercoolers, air conditioning unit heat exchangers, aircraft engine coolers, and numerous other uses. These applications most often involve gas-liquid service.

The "job" of the heat exchanger will be defined as transfering a specified amount of heat between two fluids at given flow rates (W) and with specified amounts of pumping power on each side. The following values are specified:

W_1, T_{in_1}, T_{out_1}, ΔP_1, P_{in_1}, D_{1h}, Properties_1

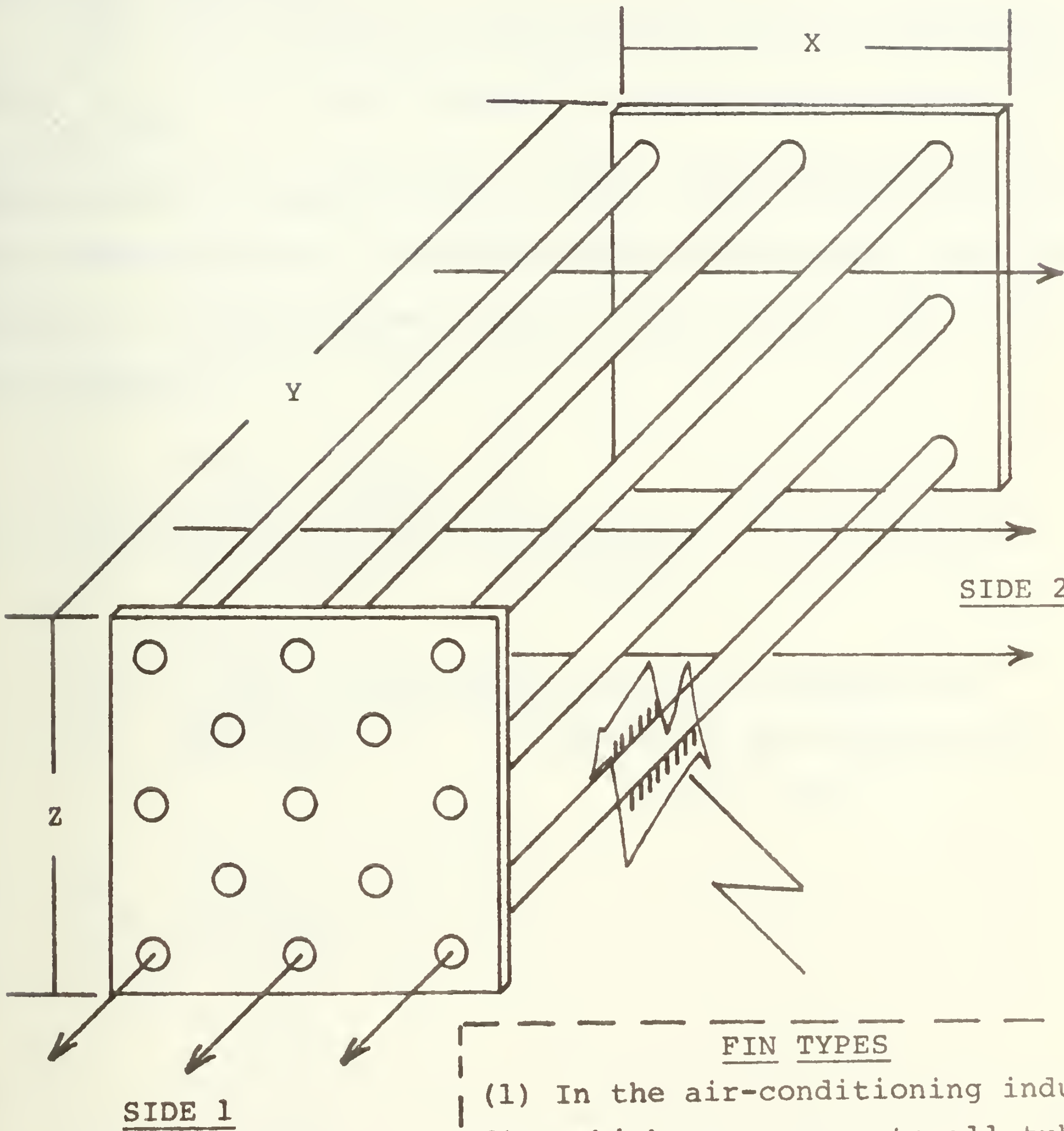
W_2, T_{in_2}, T_{out_2}, ΔP_2, P_{in_2}, D_{2h}, Properties_2

Subscript 1 will refer to the hotter fluid inside the smooth tubes and subscript 2 to the colder fluid flowing over finned tube banks.

While core pressure drop accounts for by far the greatest percentage of the total pressure drop, entrance and exit losses would have to be included in the final design. [Ref 1 Chapter 4]

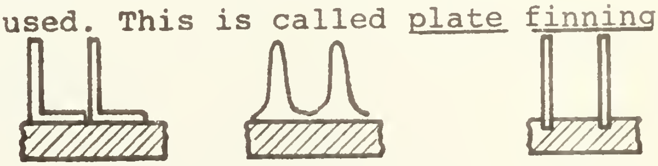
Figure A-1 shows the heat exchanger arrangement.

FIGURE A-1



FIN TYPES

(1) In the air-conditioning industry, fins which are common to all tubes in the bundle as distinct from having each tube separately finned are often used. This is called plate finning.



(2) L-fins (3) Extruded (4) G-fins

The Basic Equations

The first portion of the procedure involves determination of A_{c_1}, A_{c_2}, X, Y, and Z based on the specified values of ΔP_1 and ΔP_2 and an assumed value of RE_2. Then based on heat transfer considerations, it is determined if the proper heat balance exists. If not, another value of RE_2 is assumed and the process is repeated.

$$\beta_1 = \frac{4\, A_{c_1}}{D_{1h}\, A_{fr_1}} = \left[\frac{\pi\, D_{1h}}{F\, b}\right]_{\text{for round tubes}} \qquad D_{1h} = D_{inside} \tag{1}$$

$$\beta_2 = \frac{4\, A_{c_2}}{D_{2h}\, A_{fr_2}} = \text{A specified geometric constant for surface 2; See Figure A-II } [D_{2h} = 4r_h] \tag{2}$$

$$\frac{A_{c_1}}{A_{fr_1}} = \beta_1 \frac{D_{1h}}{4} = K_1 \tag{3}$$

$$\frac{A_{c_2}}{A_{fr_2}} = \beta_2 \frac{D_{2h}}{4} = K_2 \tag{4}$$

$$A_{c_1} = [\beta_1 \frac{D_{1h}}{4}]\ [XZ] = K_1\ X\ Z \tag{5}$$

$$A_{c_2} = [\beta_2 \frac{D_{2h}}{4}] \; [YZ] = K_2 \; Y \; Z \tag{6}$$

$$G = \frac{W}{A_c} \tag{7}$$

$$\frac{G_1}{G_2} = \frac{[W_1/A_{c_1}]}{[W_2/A_{c_2}]} = \frac{[W_1] \; [A_{c_2}]}{[W_2] \; [A_{c_1}]} \tag{8}$$

From equations (5) and (6):

$$\frac{G_1}{G_2} = \frac{[W_1] \; [K_2 \; YZ]}{[W_2] \; [K_1 \; XZ]} = K_3 \; \frac{Y}{Z} \tag{9}$$

where $K_3 = \frac{W_1 \; K_2}{W_2 \; K_1}$

$$\Delta P = \frac{4 \; f \; L \; G^2}{D_{1h} \; 2 \; g_o \; \rho_m} \tag{10}$$

where L is the flow length:

Side 1: $L = Y$

Side 2: $L = X$

$$\frac{G_1^2}{G_2^2} = \frac{[\Delta P_1 \ D_{1h} \ \rho_{m1}] \ [f_2] \ [X]}{[\Delta P_2 \ D_{2h} \ \rho_{m2}] \ [f_1] \ [Y]} \tag{11}$$

$$\text{Define } K_4 = \frac{[\Delta P_1 \ D_{1h} \ \rho_{m1}]}{[\Delta P_2 \ D_{2h} \ \rho_{m2}]}$$

From Equations (9) and (11):

$$\frac{G_1^2}{G_2^2} = K_4 \ \frac{[f_2] \ [X]}{[f_1] \ [Y]} = K_3^2 \ \frac{[Y]^2}{[X]^2} \tag{12}$$

or

$$\frac{f_2}{f_1} = \frac{K_3^2 \ [Y]^3}{K_4 \ [X]^3} = K_5 \ \frac{[Y]^3}{[X]^3} \tag{13}$$

$$\text{where } K_5 = \frac{K_3^2}{K_4}$$

Eliminate [Y/X] with equations (9) and (13):

$$\frac{f_2}{f_1} = \frac{[G_1]^3}{[G_2]^3} \ \frac{[K_5]}{[K_3]^3} = K_6 \frac{[G_1]^3}{[G_2]^3} \tag{14}$$

$$\text{where } K_6 = \frac{K_5}{K_3^3} = \frac{\Delta P_2 \ \rho_2 \ \beta_1 \ W_2}{\Delta P_1 \ \rho_1 \ \beta_2 \ W_1}$$

$$G = \frac{RE \ \mu}{D_h}$$

Substitution in equation (14) yields:

$$\frac{f_2}{f_1} = K_6 \ \frac{[RE_1^3 \ \mu_1^3 \ D_{2h}^3 \ ]}{[RE_2^3 \ \mu_2^3 \ D_{1h}^3 \ ]} \tag{15}$$

$$RE_1^3 \ f_1 = \frac{f_2 \ RE_2^3 \ \mu_2^3 \ D_{1h}^3}{K_6 \ \mu_1^3 \ D_{2h}^3} \tag{16}$$

Solving for RE_1:

$$RE_1 = \left[\frac{f_2 \ RE_2^3 \ \mu_2^3 \ D_{1h}^3}{f_1 \ K_6 \ \mu_1^3 \ D_{2h}^3}\right]^{0.333} \tag{16a}$$

To begin the design procedure, a value of RE_2 is assumed. Use the f_2 verses RE plot for side 2 to determine f_2. A sample data plot from reference [2] along with the associated geometric parameters describing a typical side 2 flow arrangement is shown in Figure A-II.

FIGURE A-II

SAMPLE DATA PLOT FOR SURFACE 2

[REF 2]

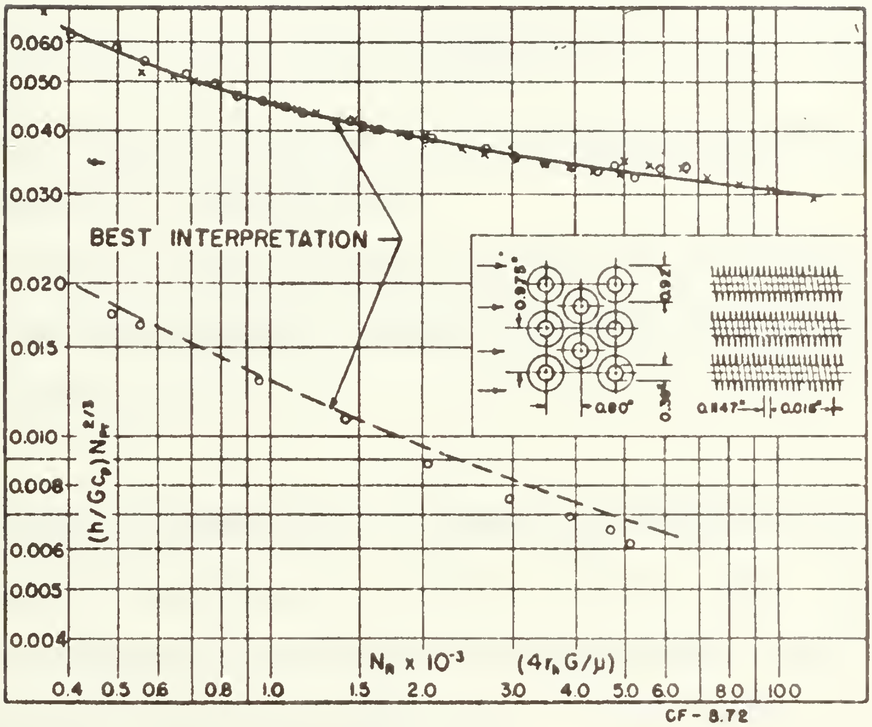


FIG. 93

FINNED CIRCULAR TUBES
SURFACE CF - 8.72

Tube outside diameter - 0.38 in.
Fin pitch - 8.72 per inch
Flow passage hydraulic diameter - $4r_h$=0.01288 ft.
Fin thickness (average)* - 0.018 in.
Free-flow area/frontal area - σ =0.524
Heat transfer area/total volume - α =163 $ft.^2/ft.^3$
Fin area/total area - 0.910

Note: Experimental uncertainty for heat transfer results possibly somewhat greater than the nominal ±5% quoted for the other surfaces because of the necessity of estimating a contact resistance in the bi-metal tubes.

* - Fins slightly tapered.

It should be noted that the friction factor verses RE plot for flow over finned tube banks can be closely approximated by a straight line on a Log-Log plot. This implies that f_2 can be accurately represented by an equation of the form:

$$f_2 = A_f[RE]^{B_f}$$

Table I list the linearized equations of f_2 [and j_2] for Figures 92-101 of reference [2].

This simplification allows for ease in programming as demonstrated in the suggested design procedure flow diagram outlined in Table II.

For exact calculations, f_1 should be read from the f_1 verses RE plot for the specific tubular surface under investigation, if such data is available. From this plot for surface 1, it is possible to determine RE_1 and f_1 to satisfy equation (16) or (16a) once a value of RE_2 has been assumed.

If, after assuming a value of RE_2, the required value of f_1 and RE_1 lay outside the range of data for surface 1, a new value of RE_2 must be assumed.

If the required data for surface 1 is not available, friction factor for turbulent flow inside smooth tubes is well correlated and given by Nikuradse as:

$$\frac{1}{\sqrt{f}} = 4.0 \, \text{Log}_{10}[RE \sqrt{f}] - 0.40 \qquad (17)$$

The transition from laminar to turbulent flow inside smooth tubes occurs between Reynolds number of 2300 and 4000.

Excellent linear approximations of the Nikuradse relation are given by:

$$f_1 \simeq 0.0791\ RE^{-0.25} \qquad 2300 < RE < 20{,}000 \tag{18}$$

$$f_1 \simeq 0.0460\ RE^{-0.20} \qquad 20{,}000 < RE < 10^7 \tag{19}$$

In a crossflow finned tubular heat exchanger the dominant heat transfer resistance will probably be the gas outside the tubes flowing over the fins (side 2). It is for this reason that high accuracy for the f (and j) prediction of the fluid on side 1 is not critical.

Note that equation (16a), using the linear approximation for f_1 given by equation (18) may be rewritten as:

$$RE_1 = \left[\frac{f_2\ RE_2^3\ \mu_2^3\ D_{1h}^3}{0.0791\ K_6\ \mu_1^3\ D_{2h}^3}\right]^{.3636} \tag{20}$$

Having satisfied either equation (16) using f_1 verses RE_1 data for the specific tubular surface under investigation, OR equation (20) using the specified linear approximation for turbulent flow inside smooth tubes, length dimensions X and Y may be calculated from the relations given in equation (10):

$$X = \frac{\Delta P_2 \; D_{2h}^3 \; g_o \; \rho_2}{2 \; f_2 \; RE_2^2 \; \mu_2^2} \tag{22}$$

$$Y = \frac{\Delta P_1 \; D_{1h}^3 \; g_o \; \rho_1}{2 \; f_1 \; RE_1^2 \; \mu_1^2} \tag{23}$$

Length Z may be calculated from equations (5) and (7):

$$Z = \frac{4 \; W_1}{\beta_1 \; RE_1 \; \mu_1 \; X} \tag{24}$$

The number of tubes can be determined from:

$$N = \frac{X \; Z}{F \; b} = \frac{W_1 \; D_{1h}}{RE_1 \; \mu_1 \; A_{c_1}} \tag{25}$$

As was the case for friction factor, j_1 should be read from the j_1 verses RE plot for the specific tubular surface under investigation, if available.

In the event that the actual heat transfer data for side 1 is not available, correlations can be used without introducing significant deviations as explained earlier.

The Colburn correlation for forced-convection, turbulent flow in tubes is widely accepted:

$$j_1 = 0.023\ RE^{-0.20} \tag{27}$$

The heat transfer coefficient [h] may be defined as:

$$h_1 = \frac{j_1\ C_{p1}\ \mu_1\ RE_1}{Pr_1^{.667}\ D_{1h}} \tag{28}$$

Or using the Colburn correlation expressed by equation (27):

$$h_1 = \frac{0.023\ RE^{0.80}\ C_{p1}\ \mu_1}{Pr_1^{.667}\ D_{1h}} \tag{28a}$$

The value of j_2 should be read from the j_2 verses RE plot for surface 2. As was the case for f_2, the Colburn modulus can be expressed in an equation of the form:

$$j_2 = A_j\ RE^{B_j}$$

Table I presents the linearized equations of j_2 for Figures 92-101 of reference [2].

$$h_2 = \frac{j_2\ C_{p2}\ \mu_2\ RE_2}{P_r^{.667}\ D_{2h}} \tag{29}$$

Refering to Figure A-III, calculate the appropriate value of fin and surface efficiency for side 2:

$$\frac{1}{AU} = \frac{1}{A_{T_1}\ U_1} = \frac{1}{A_{T_2}\ U_2} = \frac{1}{A_{T_1}\ h_1} + \frac{1}{A_{T_2}\ \eta_{o_2}\ h_2} \tag{33}$$

where:

$$A_{T_1} = \alpha_1\ XYZ$$

$$A_{T_2} = \alpha_2\ XYZ$$

If $C_1 < C_2$, $C_{min} = C_1$, $C_{min}/C_{max} = C_1/C_2$

$$\varepsilon = \frac{T_{in_1} - T_{out_1}}{T_{in_1} - T_{in_2}} \tag{34a}$$

If $C_2 < C_1$, $C_{min} = C_2$, $C_{min}/C_{max} = C_2/C_1$

$$\varepsilon = \frac{T_{out_2} - T_{in_2}}{T_{in_1} - T_{in_2}} \tag{34b}$$

FIGURE A-III

Compute fin efficiency assuming a Longitudinal fin of Rectangular Profile:

$$\eta_{f-L} = \frac{\tanh m\ell}{m\ell} \qquad (30)$$

where:

$$m = \left[\frac{2\, h_2}{k_m \delta}\right]^{.50}$$

What geometric shape describes the fin being employed on the tube surface?

Radial Fin of Rectangular Profile	Tapered (Hyperbolic) Radial fin	Rectangular Profile fin sheet [see below for ℓ defn]
$\Delta\eta_{correction} = C_\Delta + D_\Delta \ln(\eta_{f-L})$ where: $C_\Delta = -.00259 - .0084 \ln \frac{r_o}{r_e}$ $D_\Delta = -.02426 - .4521 \ln \frac{r_o}{r_e}$ [Ref 3 page 264]	$\Delta\eta_{correction} = 0$ [Ref 4 pg.3-116]	$\Delta\eta_{correction} =$ Same as Radial Fin of Rectangular Profile using r_e^* (staggered bank) $r_e^* = \left[\frac{2\sqrt{3}}{\pi}\right]^{.50} \frac{X_d\, D_o}{2}$ $\ell = r_e^* - r_o$ [Ref 4 pg.3-116]

$$\eta_{f\text{-corrected}} = \eta_{f-L} - \Delta\eta_{correction} \qquad (31)$$

Compute Surface η:

$$\eta_o = 1 - \frac{A_{fin}}{A_T}[1 - \eta_{f\text{-corrected}}] \qquad (32)$$

also

$$NTU = \frac{A\ U}{C_{min}} \tag{35}$$

If side 2 is in the form of rectangular fin sheets, the gas flowing over the tubes and the fluid in the tubes are considered UNMIXED. Using Figure A-IV with C_{min}/C_{max} and the magnitude of ε calculated from equation (34a or 34b) read the magnitude of NTU required to produce this effectiveness. Find $[NTU_{required}][C_{min}]$ and compare with the results of equation (33). If the equation is not balanced, a different value of RE_2 must be assumed and the procedure repeated.

If side 2 is in the form of low fins, the gas flowing over the tubes is considered MIXED while the fluid in the tubes remains UNMIXED. The procedure outlined above is used with Figure A-V to find the NTU required to produce the calculated effectiveness in this case. An alternative to using Figure A-V is to calculate the NTU requirement from equation (14) or (15) of reference [2].

Table II presents the entire design procedure in a flow diagram.

FIGURE A-IV

[BOTH FLUIDS UNMIXED]

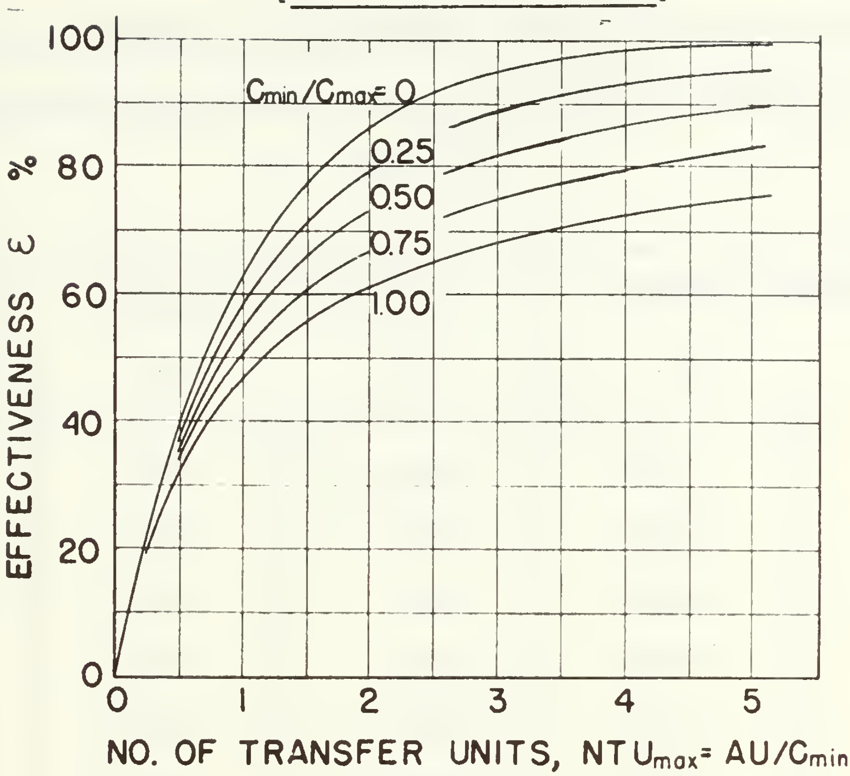


FIGURE A-V

[ONE FLUID MIXED; ONE FLUID UNMIXED]

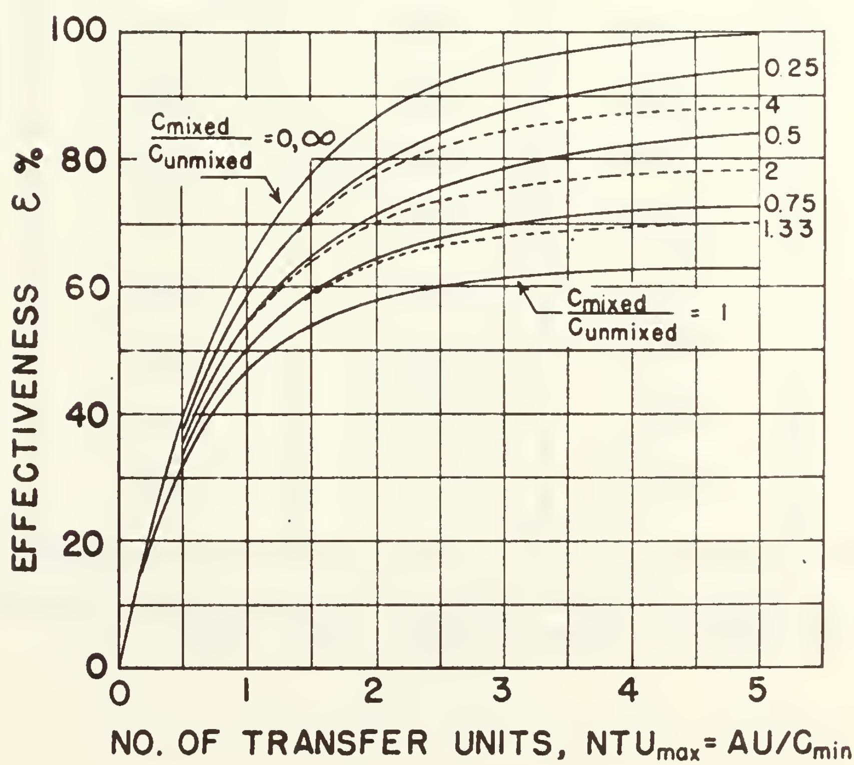


TABLE I

LINEARIZED EQUATIONS OF THE FORM A $[RE]^{B}$ FOR FRICTION FACTOR & COLBURN MODULUS FOR FINNED TUBULAR SURFACES PRESENTED IN REFERENCE [2]

FIGURE	FRICTION FACTOR [f]		COLBURN MODULUS [j]	
	A_f	B_f	A_j	B_j
92	.1781	-.1954	.3311	-.4804
93	.1862	-.2016	.2262	-.4127
94	.2377	-.2192	.2254	-.4027
96	.2646	-.2559	.1461	-.3418
97A	.2319	-.2454	.1541	-.3596
97B	.5416	-.2676	.2163	-.3666
98A	.2263	-.2707	.0842	-.3273
98B	.2905	-.2531	.0947	-.3239
98C	.3850	-.2462	.1023	-.3137
98D	.2173	-.2590	.0813	-.3264
98E	.3020	-.2607	.0915	-.3288
99A	.1542	-.2478	.1432	-.3726
99B	.3599	-.2563	.2027	-.3787
100	.1223	-.2173	.1650	-.3986
101	.0966	-.2270	.0940	-.3458

NOTE: These equations are written for the range:
$1000 < RE <$ upper limit of data IAW Ref [2]

CROSSFLOW FINNED TUBULAR HEAT EXCHANGER DESIGN PROCEDURE

INPUT

Linearized equation for f and j of the form $A[RE]^B$ for the enhanced surface: (subscript 2)

$A_{f(e)}$; $B_{f(e)}$; $A_{j(e)}$; $B_{j(e)}$

Geometric Factors

D_{1h}, D_{2h}, β_1, β_2, A_{fin}/A_T, ℓ, δ

Design Constraints:

W_1, T_{in_1}, T_{out_1}, ΔP_1, P_{in_1}

W_2, T_{in_2}, T_{out_2}, ΔP_2, P_{in_2}

Other:

Properties of the two working fluids, fin material [k] and shape, constants used to correct fin efficiency

Subscripts:

1 Hotter fluid in Smooth Tubes

2 Colder fluid flowing over finned tube banks

$$K_6 = \frac{\Delta P_2 \; \rho_2 \; \beta_1 \; W_2}{\Delta P_1 \; \rho_1 \; \beta_2 \; W_1}$$

A — Assume a value for RE_2

Assumption:

$f_1 = 0.0791 \; RE_1^{-0.25}$

$j_1 = 0.0230 \; RE_1^{-0.20}$

$$RE_1 = \left[\frac{A_{f(e)} \; RE_2^{3+B_{f(e)}} \; \mu_2^3 \; D_{1h}^3}{(.0791) \; K_6 \; \mu_1^3 \; D_{2h}^3}\right]^{0.364}$$

$$X = \frac{\Delta P_2 \; D_{2h}^3 \; g_c \; \rho_2}{2 \; A_{f(e)} \; RE_2^{2+B_{f(e)}} \; \mu_2^2}$$

TABLE II

$$Z = \frac{4\,W_1}{\beta_1\ RE_1\ \mu_1\ X}$$

$$Y = \frac{\Delta P_1\ D_{1h}^3\ g_0\ \rho_2}{0.1582\ RE_1^{1.75}\ \mu_1^2}$$

$$A_{T_1} = \beta_1 XYZ$$

$$A_{T_2} = \beta_2 XYZ$$

$$h_1 = \frac{0.027\ RE_1^{+.80}\ C_{p1}\ \mu_1}{P_r^{.667}\ D_{1h}}$$

$$h_2 = \frac{A_{j(e)}\ RE^{1+B_{j(e)}}\ C_{p2}\ \mu_2}{P_r^{.667}\ D_{2h}}$$

Go To B to compute η_0

Calculate: $\dfrac{1}{h_1\ A_{T_1}}$

Calculate: $\dfrac{1}{h_2\ \eta_0\ A_{T_2}}$

$$AU_{req'd} = NTU\ C_{min}$$

<u>DOES</u>

$$\frac{1}{AU_{req'd}} \overset{?}{=} \frac{1}{h_1\ A_{T_1}} + \frac{1}{h_2\ \eta_0\ A_{T_2}}$$

<u>NO</u>

Go To A and Assume another value for RE_2

<u>YES</u>

End of design Procedure

B

Compute fin efficiency assuming a Longitudinal fin of Rectangular Profile:

$$\eta_{f-L} = \frac{\tanh m\ell}{m\ell}$$

where:

$$m = \left[\frac{2\ h_2}{k_m \delta}\right]^{.50}$$

What geometric shape describes the fin being employed on the tube surface?

Radial Fin of Rectangular Profile

$$\Delta\eta_{correction} = C_\Delta - D_\Delta \ln(\eta_{f-L})$$

where:

$$C_\Delta = -.00259 - .0084 \ln \frac{r_o}{r_e}$$

$$D_\Delta = -.02426 - .4521 \ln \frac{r_o}{r_e}$$

[Ref 3 page 264]

Tapered (Hyperbolic) Radial fin

$$\Delta\eta_{correction} = 0$$

[Ref 4 page 3-116]

Rectangular Profile fin sheet [see below for ℓ defn]

$\Delta\eta_{correction} =$

Same as Radial Fin of Rectangular Profile using r_e^* (staggered bank)

$$r_e^* = \left[\frac{2\sqrt{3}}{\pi}\right]^{.50} \frac{X_d\ D_o}{2}$$

$$\ell = r_e^* - r_o$$

[Ref 4 page 3-116]

$$\eta_{f\text{-corrected}} = \eta_{f-L} - \Delta\eta_{correction}$$

Compute Surface η:

$$\eta_o = 1 - \frac{A_{fin}}{A_T}[1 - \eta_{f\text{-corrected}}]$$

TABLE III

SUGGESTED PROGRAM FOR SIZING CROSSFLOW FINNED TUBULAR HEAT EXCHANGERS FOR A GIVEN JOB IAW TABLE II*

I. Design Constraints/Geometric Constants to be stored in Memory as follows:

Primary Register			Secondary Register			Other		
	Quantity	Units		Quantity	Units		Quantity	Units
0	ΔP_2	LB/FT^2	0	$C_{P2}/P_r^{.667}$	BTU/LBM-F	A	RE_2 (assumed)	ND
1	$A_{f(e)}$	ND	1	$g_o \rho_{m2}$	LBM/HR^2-FT^2	B	0.00	
2	$B_{f(e)}$	ND	2	W_1	LBM/HR	C	0.00	
3	μ_2	LB/HR-FT	3	β_1	FT^2/FT^3	D	0.00	
4	D_{1h}	FT	4	ΔP_1	LB/FT^2	E	A_δ/A_T	ND
5	C_Δ	ND	5	$g_o \rho_{m1}$	LBM/HR^2-FT^2	I	β_2	FT^2/FT^3
6	D_Δ	ND	6	$C_{p1}/P_r^{.667}$	BTU/LBM-F			
7	W_1	LB/HR	7	$A_{j(e)}$	ND			
8	μ	LB/HR-FT	8	$\dfrac{C_{p2}\mu_2 \ell^2}{K_m P_r^{.667}\delta}$	FT			
9	D_{2h}	FT	9	$B_{j(e)}$	ND			

II. OUTPUT

RE_1, X, Z, Y, $\Delta\eta_{\text{to be applied}}$, $\eta_{\delta\text{-corrected}}$, $[NTU\ C_{min}]_{\text{cal'td}}$

*For Hewlett-Packard HP-67/97 Programmable Calculator

III. PROGRAM FOR SIZING FINNED TUBULAR HEAT EXCHANGERS FOR A GIVEN JOB*

[Table III continued]

001	*LBLA	21 11	057	X^2	53	113	P⇄S	16-51	169	STOD	35 14
002	P⇄S	16-51	058	÷	-24	114	RCL4	36 04	170	P⇄S	16-51
003	RCL1	36 01	059	RCLA	36 11	115	×	-35	171	RCL3	36 03
004	RCL3	36 03	060	2	02	116	RCL5	36 05	172	RCLC	36 13
005	×	-35	061	RCL2	36 02	117	×	-35	173	×	-35
006	RCL4	36 04	062	+	-55	118	PRTX	-14	174	STOC	35 13
007	÷	-24	063	Y^x	31	119	RCLC	36 13	175	LSTX	16-63
008	RCL5	36 05	064	÷	-24	120	×	-35	176	RCLD	36 14
009	÷	-24	065	2	02	121	STOC	35 13	177	×	-35
010	RCL2	36 02	066	÷	-24	122	RCLA	36 11	178	RCLI	36 46
011	÷	-24	067	RCL1	36 01	123	RCL9	36 09	179	×	-35
012	RCLI	36 46	068	÷	-24	124	1	01	180	STOD	35 14
013	÷	-24	069	RCL9	36 09	125	+	-55	181	RCLA	36 11
014	P⇄S	16-51	070	3	03	126	Y^x	31	182	RCL9	36 09
015	RCL0	36 00	071	Y^x	31	127	RCL7	36 07	183	1	01
016	×	-35	072	×	-35	128	×	-35	184	+	-55
017	RCL7	36 07	073	P⇄S	16-51	129	2	02	185	Y^x	31
018	×	-35	074	RCL1	36 01	130	×	-35	186	RCL7	36 07
019	STO7	35 07	075	×	-35	131	RCL8	36 08	187	×	-35
020	*LBLB	21 12	076	PRTX	-14	132	×	-35	188	RCL0	36 00
021	RCL3	36 03	077	STOC	35 13	133	P⇄S	16-51	189	×	-35
022	RCL4	36 04	078	4	04	134	RCL9	36 09	190	P⇄S	16-51
023	×	-35	079	RCL2	36 02	135	÷	-24	191	RCL3	36 03
024	RCL8	36 08	080	×	-35	136	$\sqrt{X}$	54	192	×	-35
025	÷	-24	081	RCL3	36 03	137	STOD	35 14	193	RCL9	36 09
026	RCL9	36 09	082	÷	-24	138	e^x	33	194	÷	-24
027	÷	-24	083	RCLB	36 12	139	RCLD	36 14	195	RCLD	36 14
028	3	03	084	÷	-24	140	CHS	-22	196	×	-35
029	Y^x	31	085	P⇄S	16-51	141	e^x	33	197	1/X	52
030	RCL1	36 01	086	RCL8	36 08	142	-	-45	198	STOD	35 14
031	×	-35	087	÷	-24	143	LSTX	16-63	199	RCLB	36 12
032	RCLA	36 11	088	RCLC	36 13	144	RCLD	36 14	200	.	-62
033	PRTX	-14	099	÷	-24	145	e^x	33	201	8	08
034	3	03	090	PRTX	-14	146	+	-55	202	0	00
035	RCL2	36 02	091	RCLC	36 13	147	÷	-24	203	Y^x	31
036	+	-55	092	×	-35	148	RCLD	36 14	204	.	-62
037	Y^x	31	093	STOC	35 13	149	÷	-24	205	0	00
038	×	-35	094	RCL4	36 04	150	ENT↑	-21	206	2	02
039	.	-62	095	3	03	151	LN	32	207	3	03
040	0	00	096	Y^x	31	152	RCL6	36 06	208	×	-35
041	7	07	097	RCLB	36 12	153	CHS	-22	209	RCL4	36 04
042	9	09	098	1	01	154	×	-35	210	÷	-24
043	1	01	099	.	-62	155	RCL5	36 05	211	RCL8	36 08
044	÷	-24	100	7	07	156	+	-55	212	×	-35
045	RCL7	36 07	101	5	05	157	PRTX	-14	213	P⇄S	16-51
046	÷	-24	102	Y^x	31	158	-	-45	214	RCL6	36 06
047	2	02	103	÷	-24	159	STOD	35 14	215	×	-35
048	.	-62	104	RCL8	36 08	160	PRTX	-14	216	RCLC	36 13
049	7	07	105	X^2	53	161	1	01	217	×	-35
050	5	05	106	÷	-24	162	RCLD	36 14	218	1/X	52
051	1/X	52	107	.	-62	163	-	-45	219	RCLD	36 14
052	Y^x	31	108	1	01	164	RCLE	36 15	220	+	-55
053	PRTX	-14	109	5	05	165	×	-35	221	1/X	52
054	STOB	35 12	110	6	06	166	1	01	222	PRTX	-14
055	RCL0	36 00	111	2	02	167	X⇄Y	-41	223	P⇄S	16-51
056	RCL3	36 03	112	÷	-24	168	-	-45	224	RTN	24

*For Hewlett-Packard 67/97 Programmable Calculator

IV. OPERATION

First Run: Select a Value of RE_2 and store in register A. To begin the program, press key "A"

Subsequent Runs: [As necessary until the calculated and required value of NTU C_{min} agree] Select a value of RE_2 and store in register A. To begin the program press key "B".

HEAT EXCHANGER DESIGN PROBLEM

APPENDIX IV

A. Description

Appendix III procedure can be used for sizing crossflow finned tubular heat exchangers. The following data, taken from reference [2], is used to determine the required heat-exchanger size for the given conditions.

The design is that of an intercooler for a 5000 SHP gas turbine plant.

DESIGN CONSTRAINT	SIDE I	SIDE II
Surface	Smooth Tube	Finned Tube 11.32-.737 SR [Fig 106]
D_h	.01224 FT	.01152 FT
δ	N/A	.00033 FT
β	42.1 FT^2/FT^3	270 FT^2/FT^3
A_δ/A_T	N/A	.845
ℓ	N/A	.225 IN
W	400,000 LB/HR	200,667 LB/HR
T_{in}	60 ^{0}F	260 ^{0}F
T_{out}	82 ^{0}F	80 ^{0}F
ΔP	1.554 PSI	.306 PSI
μ	2.36 LB/HR-FT	.0482 LB/HR-FT
C_p	1.0 BTU/LB-^{0}F	.2436 BTU/LB-^{0}F
ρ_m	62.38 LB/FT^3	.1755 LB/FT^3

	SIDE I	SIDE II
P_R	6.8	.70
K_m	N/A	100 BTU/HR-FT-^{0}F
F	.55 IN	N/A
b	.79 IN	N/A
A_c	.0560 IN^2 (From Fig 106)	Calculate

B. Calculations

$$\beta_1 = A_c/A_{fr}[4/D_h] = \frac{[.0560\ IN^2]\qquad [4]}{[.55\ IN\ X\ .79\ IN]\ [.01224\ FT]}$$

$$= 42.1\ FT^2/FT^3 \tag{1}$$

$$\beta_2 = 270\ FT^2/FT^3 \tag{2}$$

$$K_1 = [42.1]\ [.01224/4] = .129 \tag{3}$$

$$K_2 = [270]\ [.01152/4] = .778 \tag{4}$$

$$K_3 = \frac{[400{,}000]\ [.778]}{[200{,}667]\ [.129]} = 12.038 \tag{9}$$

$$K_4 = \frac{[1.554]\ [.01224]\ [62.38]}{[.3061]\ [.01152]\ [.1755]} = 1917.8 \tag{11}$$

$$K_5 = \frac{[12.038]^2}{[1917.8]} = .0756 \qquad (13)$$

$$K_6 = \frac{[.0756]}{[12.038]^3} = 4.333 \text{ X } 10^{-5} \qquad (14)$$

<u>CALCULATIONS FOR THE FINAL RE_2 SELECTED WILL BE SHOWN</u>:

Select $RE_2 = 5760$

From the data of Surface 2 [Figure 106] read:

$f_2 = .021$ $\qquad$ $j_2 = .0054$

In order to be strictly consistant with the design presented in Ref [2], friction factor [f_1] will be taken from Figure 29 of Ref [2], rather than employ any linear approximations suggested in the design procedure:

From Figure 29 data for surface 1 and Equation (16):

$RE_1 = 5073$ $\qquad$ $f_1 = .0072$

$$G_2 = \frac{RE_2\ \mu_2}{D_{2h}} = \frac{[5760]\ [.0482]}{[.01152]} = 24{,}100 \text{ LB/HR-FT}^2$$

$$X = \frac{[.306]\ [.01152]\ [144]\ [32.2]\ [3600]^2\ [.1755]}{[2]\ [0.021]\ [24{,}100]^2} \qquad (22)$$

$$X = 18.29 \text{ IN} \tag{22}$$

$$Y = 5.34 \text{ FT} \tag{23}$$

$$Z = \frac{[400{,}000]/978{,}127]}{[.129]\ [18.28/12]} = 2.08 \text{ FT} \tag{24}$$

$$\text{where } G_1 = \frac{[5073]\ [2.36]}{[.01224]} = 978{,}127 \text{ LB/HR-FT}^2$$

$$N = \frac{[18.29]\ [24.96]}{[0.55]\ [[0.79]} = 1051 \tag{25}$$

Once again, in order to be strictly consistant with the design presented in Ref [2], Colburn Modulus [j_1] will be taken from Figure 28 of Ref [2], rather than employ any linear approximations suggested in the design procedure:

From Figure 28 data for surface 1 @ $RE_1 = 5073$:

$j_1 = 0.00502$ (read $N_{st} = .0014$ and compute j]

$$h_1 = [0.005]\ [1.0]\ [978{,}127]\ [6.8]^{-.667}$$
$$= 1368 \text{ BTU/HR FT}^2\ {}^0\text{F} \tag{28}$$

$$h_2 = [0.0054]\ [.2436]\ [24{,}100]\ [0.700]^{-.667} \tag{29}$$
$$= 40.21 \text{ BTU/HR FT}^2\ {}^0\text{F}$$

$$m_2 = \sqrt{(2)\ (40.21) / (.00033)\ (100)}$$

$$= 49.3\ FT^{-1} \tag{30}$$

$$\eta_{\delta-L} = \frac{\text{Tanh}\ [49.3]\ [.225/12]}{[49.3]\ [.225/12]} = 0.788 \tag{30a}$$

$$\Delta\eta_{correction} = 0 \text{ [For consistancy with Ref (2) example]*} \tag{31}$$

$$\eta_o = 1 - .845[1 - .788] = 0.82 \tag{32}$$

$$\frac{1}{AU_{calculated}} = \frac{1}{[1368][711.55]} + \frac{1}{[40.21][0.82][4570.9]}$$

$$= \frac{1}{130{,}506\ BTU/HR\text{-}^0F} \tag{33}$$

In order to complete the design procedure, compute $AU_{required}$ and compare with equation (33) results:

$$[AU]_{required} = NTU\ C_{min}$$

$$C_{min} = C_{air(side\ II)} = [200{,}667]\ [.2436]$$
$$= 48{,}883\ BTU/HR\text{-}^0F$$

$$C_{max} = C_{water(side\ I)} = 400{,}000\ BTU/HR\text{-}^0F$$

$$\varepsilon = \frac{260 - 82}{260 - 60} = 0.89 \tag{34b}$$

From Figure 5 of Reference [2], for a cross-flow exchanger with both fluids UNMIXED, and $C_{min}/C_{max} = 0.122$ with $\varepsilon = 0.89$, [or from Table 4 of Ref [2] with proper interpolation] the $NTU_{req'd}$ is found to be 2.67.

$$NTU_{req'd}C_{min} = [2.67][48{,}883] = 130{,}517 \text{ BTU/HR-}{}^0\text{F} \tag{35}$$

Comparing Equation (33) with equation (35):

$$NTU_{req'd}C_{min} \stackrel{?}{=} NTU_{calculated}\ C_{min}$$

$$130{,}517 \text{ BTU/HR-}{}^0\text{F} = 130{,}506 \text{ BTU/HR-}{}^0\text{F}$$

CLOSE ENOUGH!

---

* Surface 2 is a rectangular fin sheet and, to be strictly correct, a correction should be applied to permit the use of equation (30a) as outlined in the design procedure.[See Appendix III, Table I] In order that the results of this example will be consistant with those of reference [2], a correction of 0 will be assumed pursuant to the simplifying assumptions as stated in the example of reference [2].

Thesis 171161
B242349 Baskerville
Performance ranking
of finned tubular
heat exchanger
surfaces.